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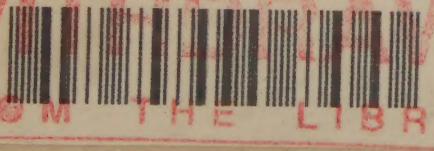

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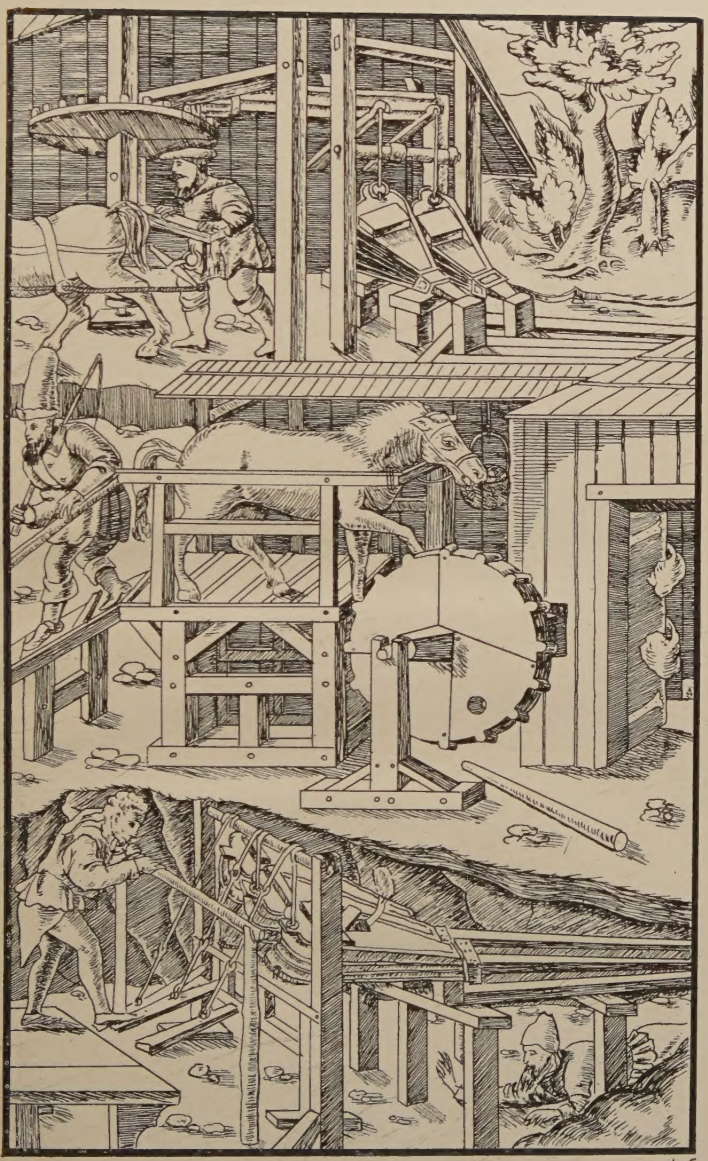

Frontispiece] MINE VENTILATION IN THE DARK AGES.
(Agricola's *De Re Metallica.*)

MINE VENTILATION

AND

VENTILATORS

BY

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With 191 Illustrations.

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PREFACE.

THE subject of Mine Ventilation and Ventilators finds a prominent place in all general works on mining, such as *A Handbook on Coal Mining*, by Herbert Hughes, and *Practical Coal Mining*, by George L. Kerr, but such general reference is not sufficient for present-day students and mining engineers. The subject requires wide knowledge concerning mine atmosphere and mine gases, temperature and humidity in mines, mine pressure and resistances, and the fundamental principles whereby ventilation of mines is accomplished, as well as a knowledge of the machinery designed for the purpose.

The author in the course of his teaching has experienced the need of a concise practical handbook dealing fully with first principles of this branch of the mining engineer's work, and, bearing in mind the great complexity of mining operations, he has endeavoured to present herein the elements of the subject and an explanation of some points seemingly obscure, together with short descriptions of machines of proved usefulness and other apparatus of most recent design, so as to aid students in acquiring the needful preparation for securing their certificates as mine-managers and fit them for responsible duties awaiting them. For purposes of reference, hygrometric tables are included at the end of the book, with the kind permission of Dr. J. W. L. Glaisher, of Cambridge.

The author gratefully acknowledges most kind assistance given by Mr. J. Ivon Graham, M.A., M.Sc., of the Research Department, the University of Birmingham; Mr. Thomas Bryson, of the Wigan Mining College; and Dr. J. W. L. Glaisher, F.R.S., of Trinity College, Cambridge. He is also greatly indebted to Dr. Hy. Briggs, of Edinburgh; Mr. H. C. Harris, of Coatbridge; Principal Penman, of the Dhanbad School of Mines, Bengal; Mr. D. E. Thomas, of Nuneaton; Mr. Joseph Parker, of the

Mining School, Cowdenbeath ; Mr. B. W. Holman and others for permission to draw upon their published works and writings.

For descriptions of machinery and for the loan of illustrations, thanks are tendered to manufacturers, acknowledgment being made in the text. Lack of skill as a draughtsman afforded the opportunity of utilising the abilities of senior students, whose kind assistance is gratefully acknowledged.

That the publication of this little book should be undertaken by Messrs. Charles Griffin & Co. Ltd., is most gratifying to the author, whose first attempt at preparing a manuscript for press has been safeguarded by their supervision. The conversion of notes prepared for *viva voce* lectures into the more exact expression needed for a text-book has proved difficult, and should errors have crept in, the author will be obliged by his attention being called to them.

M. H. H.

Coalville, Leicester,
November 1923.

CONTENTS.

CHAP.		PAGES
I.	Definitions	1–37
II.	Fundamental Laws and Theorems . .	38–62
III.	The Atmosphere and Mine Gases . .	63–91
IV.	Mine Temperatures and Humidity . .	92–123
V.	Mine Pressures and Resistances . .	124–147
VI.	The Production of Ventilation in Mining	148–208
VII.	The Laws of Mine Ventilation . . .	209–249
VIII.	The Practice of Mine Ventilation . .	250–270
IX.	Types of Centrifugal Ventilators . .	271–289
X.	The Theory of the Mine Fan . . .	290–311
	Hygrometric Tables	312–331
	Index	332

LIST OF FOLDING PLATES, ETC.

Mine ventilation in the Dark Ages . . . . . *Frontispiece*

Table X. Strata temperatures in the Coal Measures . *To face page* 94

FIG.

47. Air-cooling plant at Morro Velho Mine, Brazil . . ,, 109

139–142. Automatic airlock doors . . . . . ,, 257

143–145. Airlock at Maltby Main Colliery . . . . ,, 258

154. Ventilation of a hot and gassy seam . . . . ,, 266

155. Ventilation with downcast the winding shaft . . . ,, 268

156. Enlarged plan of pit bottom. . . . . ,, 268

157. Ventilation with six splits of air . . . . . ,, 268

158. Ventilation with six splits of air . . . . . ,, 268

159. Ventilation with five splits of air and two air crossings . ,, 268

160. Ventilation with five splits of air and four air crossings . ,, 268

161. Ventilation with four splits of air and two air crossings . ,, 268

162. Ventilation with five splits of air and four air crossings . ,, 268

163. Modern machine-developed shaft pillar (plan to be ventilated) . . . . . ,, 270

164. Panel-longwall development (plan to be ventilated) . ,, 270

166, 167. Installation of Walker fan at Blackhall Colliery . . ,, 273

NEWTON'S LAWS OF MOTION

1st Law.

Every body continues in a state of rest or uniform motion in a straight line except in so far as it is compelled by the action of force to change this state.

2nd Law.

Rate of change of motion is proportional to the impressed motive force, and takes place along the line in which that force is impressed.

3rd Law.

Action and reaction are equal and opposite; or, the mutual actions of two bodies are always equal and oppositely directed.

MINE VENTILATION AND VENTILATORS.

CHAPTER I.

DEFINITIONS.

Matter is anything which exists and of which we have physical experience, *i.e.* it is that which occupies space. Any portion of matter bounded by definite limits is called a body. Matter may exist in any of the three forms —solid, liquid, or gaseous, the last two forms comprising fluids. It cannot be created or destroyed, its form only can be changed. This we find well proven by disintegrating a known quantity of matter physically or chemically and comparing the quantities after the process; a perfect balance of the quantities in the original and final forms will result. The ultimate constitution of matter is unknown to the present age, and much investigation thereon is now progressing.

An element is a substance which cannot be split up into simpler substances, *e.g.* iron, oxygen, carbon, etc. There are about eighty-seven known elements broadly classified as metals and non-metals. This division is not perfect, because there is no definite line of demarcation between metals and non-metals; but the latter include the gaseous elements, and frequently have names ending in " -on," whilst the former are the remaining elements, and their names usually, but not always, end in " -ium."

A mixture is formed when any number of elements in any proportion are mixed together without undergoing chemical action, *e.g.* iron and sulphur mixed under ordinary conditions.

A chemical compound is formed when two or more elements combine together to form a totally different substance, *e.g.* when

two volumes of hydrogen (H) and one of oxygen (O) are exploded in contact by means of a lighted taper, water (H_2O) being formed.

Physical change occurs when a body changes its state, generally by means of heat, without changing its chemical composition, *e.g.* water to steam. In cases of physical change the change is temporary, and the body can be returned easily to its former state.

Chemical change occurs when the change of a body's state is accompanied by change of composition. Thus coal, which contains many complex compounds, is, by means of heat, split up into substances which are unlike the original, and the disintegration is irreparable. It is generally difficult, and in many cases impossible, to bring a body back to its former state after chemical change.

Mass expresses the quantity of matter in a body, and is measured by comparison with a standard body, which in this country is a piece of platinum kept at the Board of Trade Offices, Westminster, of 1 lb. avoirdupois. The usual method of measuring mass is by the beam balance or modified forms thereof. In countries using the metric system of measurement, the gram, which is the quantity of matter contained in one cubic centimetre of water when at its maximum density—at 4° C. or 39° F. approximately,—is taken as the standard of mass.

Density is the term used when stating the quantity of matter or the mass of a cubit unit of any substance. Thus two different substances, or two similar substances under different conditions of pressure and temperature, contain different quantities of matter, and that which holds the greater quantity, or mass, is said to be the denser of the two. For instance, the density of water is 62·3 approx., there being about 62·3 lbs. mass in 1 cub. ft. of water; air has a density at 0° C. and normal pressure conditions of about 0·0807 lb. per cub. ft., whilst for iron it is 480 lbs. approx.

Force is that which changes or tends to change the state of rest or motion of a body. We say "tends," because all bodies are not affected by all forces, since some forces are weak and inappreciable ones. The downward pull of a body, due to gravitation, or its weight as measured by a spring balance, is the measure of the magnitude of the force it exerts in lbs. Static forces are those which apply to bodies at rest or in a state of equilibrium, whilst those applying to moving bodies are known as dynamic forces, and the branch of mechanics dealing with

this latter is called kinetics, as distinguishing it from statics applying to the former. A force is fully defined when we know its point of application, its direction, and its magnitude. All bodies attract one another with a force which varies in accordance with the product of the masses of the bodies and inversely as the square of the distance between their centres. This may be generally stated with respect to two masses M and m, whose centres are at a distance r apart by $\frac{Mm}{r^2}$. Thus the greater the mass the greater the pull, and the greater the distance the less the pull. Now the earth being overwhelmingly greater in mass than any body at, or near, its surface, attracts all these, so that when unobstructed they fall in a straight line towards the earth's centre, and in this way the attraction of these lesser bodies for each other is obscured. For the converse reason that the distances and masses of the stars are much greater, the earth has no visible effect upon the stellar bodies.

Weight is the definition of the above force with which the earth attracts a body, and is, consequently, a measure of the mass of the body. This force is not uniform or invariable, since from the foregoing it will be seen that it depends upon the mass and the distance of the body from the earth. Whilst from the above it will be also seen that weight and mass may be measured by the same number, it is extremely important to understand the difference between mass and weight. The former represents the quantity of matter in a body, and is independent of gravity; the latter is the force exerted by the body and is directly dependent on gravity.

Specific gravity, or sp. gr., expresses the ratio of the weight of a given volume of a substance to that of an equal volume of a standard substance. In engineering and most general mining calculations water is taken as the standard; in chemistry hydrogen, being the lightest known substance, is the usual standard, whilst in most problems on ventilation air is taken as the standard unit for comparison. Sp. gr., therefore, is simply a number: *e.g.* with hydrogen as the standard or unity the sp. gr. of normal air which weighs 0·08075 lb. per cub. ft. must be 14·4, because the weight of an equal volume of hydrogen is 0·00559 lb., and $\frac{0{\cdot}08075}{0{\cdot}00559}$ expresses this ratio. When air is taken as the comparative standard base, as often in computations on fans and mine atmospheres, air at normal temperature and pressure

(N.T.P.), *i.e.* a pressure of 760 mm. of mercury and 0° C. is usually meant. Another standard favoured by mining men is that of air at 60° F. and under a pressure of 30 in. of mercury.

Pressure is the term given to the measure of a force on a given unit area, and for this reason is expressed in tons per sq. ft., lbs. per sq. inch, etc. From this it is apparent that the product of the pressure and the total area affected, stated in the proper units, denotes the total pressure on the area being discussed.

Velocity defines the rate at which a body moves with respect to other bodies. As rate implies time, the fact that velocity cannot be stated in any single dimension, like " 60 miles," will be patent; it must, of course, be stated also in what time this distance is covered, *e.g.* 60 miles per hour. When a body's velocity is referred to things at rest its absolute velocity is then defined, but when the same is stated with regard to another moving body its relative velocity is then being considered. Thus, if a train is moving at a velocity of 60 m.p.h. with respect to the earth, and another moves in the same direction at 45 m.p.h., the relative velocity of the first train to the second one is $60 - 45$, or 15 m.p.h., and if they are moving in opposite directions it is $60 + 45$ m.p.h.

A body falling freely in a vacuum, *at the end of one second*, acquires a velocity of 32·2 ft. per second, *i.e.* its average velocity throughout the second is the mean of that at the beginning and end of the second, or 16·1 ft. per second for the first second. Again, at the end of the second second its velocity is $2 \times 32{\cdot}2$ ft. per second, and after 5 seconds $5 \times 32{\cdot}2$ ft. per second, and after n seconds $n \times 32{\cdot}2$ ft. per second, and so on, increasing at 32·2 ft. per second each second.

Acceleration is the name given to the above periodic increment if additive, but if subtractive it is called deceleration or retardation. For a body falling freely under the influence of gravity the acceleration is, in London, about 32·2 ft. per second each second. It varies slightly with the latitude, and will be a little greater at the poles, since those places are nearer the centre of the earth. When this increment is regular in point of time and magnitude the acceleration is said to be uniform, just as a velocity is uniform when it is constant over equal periods of time and space. From the above it will be noted that a velocity v ft. per second is the product of the acceleration a ft. per second per second, and the time t seconds, thus $v = at$ . . . . (1) and since the distance, S, traversed is the product of the average

velocity and the time, then $S = \frac{1}{2}vt$ . . . (2)
But from (1) we get by substitution $S = \frac{1}{2}(at) \times t$
$\therefore S = \frac{1}{2}at^2$ . . . (3)
Again from (1) transformed and substituted in (3)

$$S = \tfrac{1}{2}a\left(\frac{v}{a}\right)^2$$

$$S = \frac{v^2}{2a} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or $v^2 = 2aS$ . . . . . (5)
or $v = \sqrt{2aS}$ . . . . . (6)

The same procedure holds for a body falling freely under the influence of gravity where a height, h ft., is passed through in a time, t seconds, having a final velocity v ft. per second, due to the effect of the acceleration by gravity, g ft. per second per second, thus—

$v = gt$ . . . . . (1a)
$h = \frac{1}{2}vt$ . . . . . (2a)
$h = \frac{1}{2}gt^2$ . . . . . (3a)

$$h = \tfrac{1}{2}g\left(\frac{v}{g}\right)^2 = \frac{v^2}{2g} \quad . \quad . \quad . \quad . \quad (4a)$$

hence $v^2 = 2gh$ . . . . . (5a)
or $v = \sqrt{2gh}$ . . . . . (6a)

If the body has an initial velocity of say u ft. per second previous to any of the above considerations, the general method is preserved as in the foregoing cases and only modified in respect to the addition of u, thus—

$v = u + at$ . . . . . (1b)
and $h = \frac{1}{2}(u+v) \times t$ . . . . . (2b)
hence $h = ut + \frac{1}{2}at^2$ . . . . . (3b)

and eliminating t by cross multiplying (1b) by (2b)

$$h = \frac{(v^2 - u^2)}{2a} \quad . \quad . \quad . \quad . \quad (4b)$$

$v^2 = u^2 + 2ah$ . . . . . (5b)
or $v = \sqrt{u^2 + 2ah}$ . . . . . (6b)

To produce acceleration of a body a force or forces must act on it, and if no such force exists the body remains in its former

state of rest or uniform motion as the case may be. This feature of matter, much akin to human laziness or disinclination to change the existing state of motion, is known as inertia, and has to be overcome by force if any change is desired in the motion of a body (see Newton's First Law). This expenditure of force needed to overcome inertia will be noted when one observes the effort needed to start or stop an engine; in either case the force required is greater than that needed to keep it in uniform motion, or of course at rest. In a body of mass one pound a force of 1 lb. weight acts upon it, producing, if free to fall, an acceleration of 32·2 ft. per second per second (32·2 ft. per second per second, or 32·2 ft. per second^2 for short). Now, if we could take the body to such a height that its weight is halved, its mass of course remaining the same, the acceleration produced would be $\frac{32\cdot2}{2}$ ft. per second^2, and if we could further reduce its weight to $\frac{1}{32\cdot2}$ lb. the acceleration produced would be 1 ft. per second^2. The inertia in each case here is 1 lb., $\frac{1}{2}$ lb., and $\frac{1}{32\cdot2}$ lb. respectively.

On the other hand, if the mass could be increased to 32·2 lbs., still keeping the weight at 1 lb., the acceleration would be 1 ft. per second^2. Hence the law "*the force required to produce in a body a given acceleration is proportional to the product of the mass of the body and the required acceleration.*" Thence, since a force of 1 lb. produces an acceleration of 32·2 ft. per second^2, we can now calculate the force P needed to give an acceleration a ft. per second^2 to a mass m, thus—

$$P = ma, \text{ lbs. wt.} = \frac{Wa}{g} = \frac{Wv}{gt} \quad . \quad . \quad (7)$$

where W is the weight as distinguished from the mass m. The poundal is the unit of force, being that force required to give an acceleration of 1 ft. per second^2 to a mass of 1 lb.; obviously it is got by a force of $\frac{1}{32\cdot2}$ lb. wt. acting under such conditions.

Work is said to be done when a force acts through a distance, and in so doing it overcomes resistance. In such cases work may be done either *by* or *on* the force. The British unit employed is the "foot-pound" (ft.-lb.), which is the work done when a force or weight of 1 lb. acts through a distance of 1 ft. That is,

if a force P has acted through the *entire* distance h, the work done, u, is

$$u = Ph \text{ ft.-lbs.} \quad . \quad . \quad . \quad . \quad (8)$$

Fig. 1.

But if the point of application of the force has been displaced, due to the action of other forces or bodies, through an angle @ from the line of action of the force the component of the force is $h \cos @$, thus modifying the work done to

$$u = Ph \times \cos @ \quad . \quad . \quad . \quad . \quad (9)$$

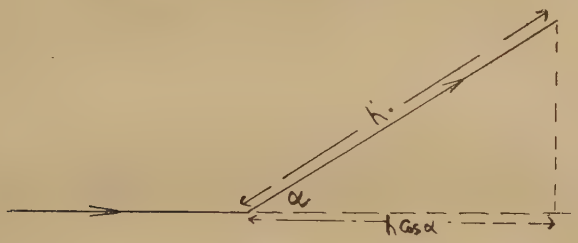

Fig. 2.

It is very important to note the above fact, which recurs throughout all physical problems involving forces in mine ventilation, especially in the design of fan blades. From this it will be seen that work is equal to force multiplied by distance, which distance is measured always in the direction of the force, to which the position of the point of application must, in all cases, be resolved. 33,000 ft.-lbs. of work per minute, or 550 ft.-lbs. per second, constitute 1 horse-power, or h.p., though this really is about 40 per cent. more than a normal horse can achieve continuously.

Energy is the capacity for doing work. If work is stored in a body electrically, mechanically, hydraulically, or by virtue of its position or state of motion, the body is said to possess energy. Mine air in motion has acquired sufficient energy to overcome

effectively the resistance of the mine; the mine air expends its energy in so doing. If work, as in the case of the resistances of the underground airways above, is done against the force moving the air, the air loses energy, causing, if there is insufficient energy, stagnancy and the accumulation of deleterious gases. If, on the other hand, work is done on the ventilating force by, say, an auxiliary or booster fan in the workings, or any other force acting on the initial ventilating force, it gains energy. The loss or gain of energy will be readily deducible in each case, and in all cases will establish the following fact: "Energy supplied equals energy doing useful work plus energy lost in overcoming friction." This fact holds, regardless of what form or state the original energy is transformed into, and is known as the principle of the conservation of energy, just as the principle of the indestructibility of matter is called the conservation of matter. Therefore neither matter nor energy can be destroyed or created by man, they can only be transformed from an already existing state or form into another state or form. The root nature of energy, like that of matter, is only very vaguely understood at present, and the quest of knowledge in this direction forms one of the most engrossing aspects of modern endeavour. The broadest, but by no means the entire, classification of the forms of energy is that of potential energy and kinetic energy. Potential energy is that possessed by a body by reason of its position, from which, when released, it is transformed, provided we assume no friction, into an equal amount of motion energy or kinetic energy. Potential energy, or the energy of position, can be well appreciated if we imagine the weight of heavier cold air in a downcast shaft to be constantly tending to press the warmer and therefore lighter mine air up the upcast shaft. Here we have a mass of atmospheric air which was formerly quiescent, and therefore only possessed of potential energy, undergoing the change into kinetic energy and so performing useful work. Be it noted that potential energy (P.E.) is quite as useful and important as kinetic energy (K.E.), since it is a reserve of what might be considered dormant energy. Another example is the P.E. of a coiled spring.

A weight, W lbs., at a height, h ft., above the ground, possesses $W \times h$ ft.-lbs. of P.E., and if allowed to fall will acquire a velocity

by (6a) $\quad v = \sqrt{2gh}$.

or by (5a) $\quad v^2 = 2gh$

whence by (4a) $$h = \frac{v^2}{2g}$$

and in doing so has lost all its P.E., converted now into K.E.

P.E. lost $= Wh$ — K.E. gained, and, substituting the value of h above, $Wh = W \times \frac{v^2}{2g}$ ft.-lbs.

Thus kinetic energy $= W\frac{v^2}{2g}$ ft.-lbs. . . . . (10)

This result is so important that a discussion of it from another viewpoint will aid us in this expression of work and energy. By modifying (7) we see that the acceleration a ft. per second^2 of a body of W lbs. until it has acquired a velocity v ft. per second due to a force P lbs., is obtained from the quotient of the accelerating force and the mass, thus—

$$a = \frac{P}{\frac{W}{g}} \text{ or } \frac{32{\cdot}2P}{W} \text{ ft. per second}^2;$$

again, from (1a) the time taken to attain this velocity is $t = \frac{v}{g}$, and the distance moved through at the average velocity of $\frac{v}{2}$ ft. per second is by former definition $= \frac{v}{2} \times t$. We are also able now to state t in another form, *i.e.* by substituting in (7) the value of "a" from (1) thus—

by (1) $$a = \frac{v}{t}$$

and by (7) $$P = \frac{Wa}{g} = \frac{Wv}{gt} \therefore t = \frac{Wv}{gP},$$

hence the distance traversed is $\frac{v}{2} \times \frac{Wv}{gP} = \frac{Wv^2}{2gP}$,

and the K.E. or work done = distance × force =

$$\frac{Wv^2}{2gP} \times P = \frac{Wv^2}{2g} \text{ ft.-lbs. as before} \quad . \quad . \quad (10)$$

This applies to any body starting from rest.

From this result it will be evident that mass and weight are interchangeable and synonymous as long as we are comparing

the weight of one body with the weight of another. Wherefore the type statement for the K.E. is $\frac{1}{2}mv^2$ when speaking in terms of mass, and $\frac{Wv^2}{2g}$ when dealing in foot-pounds.

Among other aspects of energy may be mentioned strain due to elasticity, and also gravitation as types of P.E.; chemical action, light, and heat are examples of K.E., whilst vibratory properties and the intermolecular nature of matter are instances of P.E. and K.E. combined.

Angular Velocity.—If a body moves in a circular path with a uniform velocity v ft. per second at a distance r ft. about a fixed centre, the time t seconds taken for a complete circuit is

$$t = \frac{2\pi r}{v} \text{ seconds} \quad . \quad . \quad . \quad (11)$$

Now the angle subtended at the centre of a circle by an arc equal in length to the radius is called a *radian*, and there are obviously 2π radians in a complete revolution of the circle. So the time taken to traverse one radian will be

$$\frac{\text{distance}}{\text{velocity}} = \frac{r}{v} \text{ seconds} \quad . \quad . \quad . \quad (12)$$

and the angular velocity ω radians will be

$$\omega = \frac{v}{r} \text{ radians per second} \quad . \quad . \quad (13)$$

Moment of Inertia.—The rotational K.E. of a body of weight W lbs. with a velocity v ft. per second, is given by considering the above general assertion, $\frac{Wv^2}{2g}$ or $\frac{1}{2}mv^2$, when the body has an angular velocity, ω radians per second at a radius r ft., about a fixed centre. This velocity is then, $r \times \omega$, thus making for a weight W, a rotational K.E. of

$$\frac{W}{2g}(r\omega)^2 \text{ or } \frac{1}{2}\frac{W}{g}r^2\omega^2 \text{ ft.-lbs.} \quad . \quad . \quad . \quad (14)$$

The sum of the products of all the parts making up the rotating system and of the order $\frac{W_1}{g}, \frac{W_2}{g}, \frac{W_3}{g}$, etc. . . . and the squares of their respective distances $r_1^2, r_2^2, r_3^2, \ldots$ etc., from the centre of rotation, is called the moment of inertia of the whole body round the axis of rotation. It will be readily understood if we

are considering the forces in a ventilating machine, for example, that no ventilator is a thin cylinder turning about its axis of rotation, but is in reality a number of collected parts W_1, W_2, W_3, etc., all of different weights and situated at different distances, r_1, r_2, r_3, etc., from the centre. Consequently only the summation of these quantities arranged in the order of (14) above can effectively denote the forces to be considered in designing and equipping a satisfactory ventilating appliance.

The moment of inertia is usually denoted as I, whereby the awkward algebraical expression which the said method would produce, *i.e.*

$$\text{total K.E.} = \frac{W_1}{2g} r_1^2 \omega^2 + \frac{W_2}{2g} r_2^2 \omega^2 + \frac{W_3}{2g} r_3^2 \omega^2 \quad . \quad (15)$$

or, as it is often stated,

$$\text{total K.E.} = \frac{\omega^2}{2g} [W_1 r_1^2 + W_2 r_2^2 + W_3 r_3^2 + \ldots] \quad . \quad (16)$$

may now be more conveniently expressed by

$$\text{total K.E.} = \tfrac{1}{2} I \omega^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

which is known as the body's kinetic energy of rotation.

Circular Motion.—Now imagine two bodies A and B revolving at a fixed angular distance of, say, 60° from one another in a

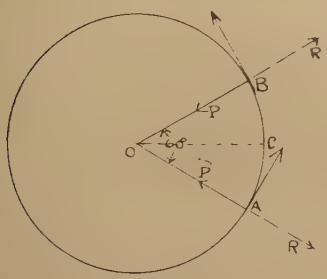

Fig. 3.

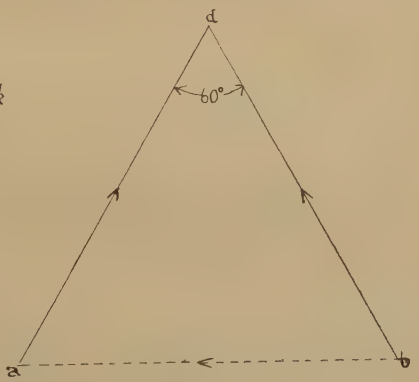

Fig. 4.

circle about a centre O, and so having the same velocities and accelerations, but of course not in the same directions (see arrows, figs. 3 and 4).

The natural tendency will be for the bodies to fly off at a tangent as shown by the arrows at A and B in fig. 3, but since A has the same circular velocity in any part of the revolution it follows that it maintains this uniform velocity against a force which is constantly changing its direction. In other words, it must be accelerating uniformly toward the centre O during its motion; this will be seen better perhaps if the successive positions of A or B are projected on to the midway line O C. This inward pull P is exerted by anyone whirling a sling. The inherent tendency of a body to move in one straight line and velocity, or its inertia, gives to the body a proneness to resist the pull at the circumference, which pull is exerted at the centre, as in the case of the sling. This inward force is called the centripetal force, and it restrains the body's motion to a circular path, while the outward pull R at A, which is equal and opposite to P, is called the centrifugal force. In the case of a sling, P ceases when the limb of the sling is released and R becomes apparent immediately in that the body flies off in the direction, and at the speed, it had on becoming detached. There is no outward force on the body, the body rather exerts an outward pull equal and opposite to P on the string or arm of the moving object.

Reverting to fig. 3, and paralleling the velocities off in direction and magnitude (say 3 ft. per second as velocity), we obtain the graphical representation of the forces or vector diagram, fig. 4, which is an equilateral triangle, since the velocities are equal, and acting at 60° apart. In this *ad*, *db*, and *ba* are all equal, also the chord *ab* is equal to the radius, since the angle is 60°. A will take one-sixth of the time taken for a complete revolution. This by (11) is $t = \frac{1}{6} \times \frac{2\pi r}{v} = \frac{\pi r}{3v}$ seconds, and the change of velocity per second, *i.e.* the average acceleration, is

$$\frac{\text{Change of velocity}}{\text{time}} = \frac{v}{t} = \frac{v}{\frac{\pi r}{3v}} = \frac{3v^2}{\pi r} = \frac{0{\cdot}956v^2}{r}$$

feet per second per second. This is nearly the acceleration at C midway between A and B, and is in the direction from C to O (toward the centre). Again (see figs. 5 and 6), if the angle be taken more acute still, say 6° instead of 60°, we approach nearer to unity as the coefficient of $\frac{v^2}{r}$. In this case the chord or vector

base with unit radius is 0·10467, and the time for $\frac{1}{60}$th of a circle, $t = \frac{1}{60} \times \frac{2\pi r}{v} = \frac{\pi r}{30v}$ seconds, from which, by the above method, we obtain the mean acceleration of $\frac{9{\cdot}549\, v^2}{r} \times 0{\cdot}10467$ or $0{\cdot}999\frac{v^2}{r}$ ft. per second^2. The value of the acceleration is found to be

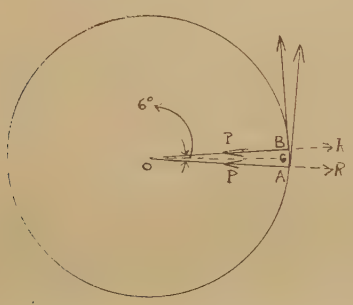

Fig. 5.

Fig. 6.

greater and greater as we take the angle smaller and smaller, until finally we obtain $\frac{v^2}{r}$ ft. per second^2 . . . . (18)

as the acceleration at any point of a body moving uniformly in a circle. Now since $v = \omega r$, by (13), etc., the acceleration may be written $a = \frac{(\omega r)^2}{r} = \omega^2 r$, ft. per second^2 . . . (19)

Further, by (7) $P = \frac{Wa}{g}$,

whence by (18) $P = \frac{Wv^2}{gr}$, lbs. = centrifugal force . . . (20)

The expression (20) is often stated thus, $P = \frac{W}{g}\omega^2 r$, which may be obtained from the product of the mass and the acceleration $\omega^2 r$, thus—

$$\frac{W}{g} \times \frac{v^2}{r} = \frac{Wv^2}{gr} \text{ or by (13) } \frac{W\omega^2 r^2}{gr} = \frac{W}{g}\omega^2 r \text{ . . . (21)}$$

Examining (20), $P = \frac{Wv^2}{gr}$, for the centrifugal force, we see that the smaller the fan (in diameter) the greater the centrifugal force. This force varies as the square of the velocity and inversely as the radius, other things being equal. Thus a 100-in. fan will impart $\frac{25 \times 12}{100}$, or three times as much centrifugal force to a given unit of air as a fan 25 ft. in diameter under the same conditions of work, *i.e.* the same tip speed. Small fans for laboratories, etc., often, for the above reasons, appear to give incredible results, whereas it is only a matter of size. Therefore at a given tangential speed the smaller fan develops the greater centrifugal force, which in a large measure explains the modern bias for small fans moving quickly in order to have the same tangential velocity as larger fans, and also maintain the above advantage over them.

Power is the time-rate of doing work, *i.e.* it is the amount of work any agent (engine, man, etc.) performs per unit of time, *e.g.* if a weight of 8000 lbs. of air per minute is extracted from a mine 1000 ft. deep, 8000×1000 ft.-lbs. of work is accomplished by the ventilator per minute. Knowing that 33,000 ft.-lbs. per minute, or 550 ft.-lbs. per second, comprise 1 h.p., then the machine producing the ventilation has developed $\frac{8000 \times 1000}{33000} = 242{\cdot}4$ h.p. in useful work. When work is expressed barely in h.p. the time is understood from the factors by which the h.p. is obtained.

It is often convenient to express the power as the total quantity of units exerting pressure for a given time. Thus if p is the pressure in lbs. exerted by the unit, say 1 cub. ft. of air, on a square foot, and Q, cub. ft., is the quantity of such units acting per minute, then

$$\text{h.p.} = \frac{Qp}{33000} \quad . \quad . \quad . \quad . \quad (22)$$

Again, since the quantity of air is the volume of the space occupied by it, *i.e.* the area of the mean cross-section of an airway multiplied by its length, or the cross-section "a" multiplied by the distance travelled by the air in a unit of time (usually a minute), in other words, multiplied by the velocity, v ft. per minute, through the section, then

$$Q = av \quad . \quad . \quad . \quad . \quad (23)$$

from which $$v = \frac{Q}{a} \quad . \quad . \quad . \quad . \quad (24)$$

and $$a = \frac{Q}{v} \quad . \quad . \quad . \quad . \quad (25)$$

We may now state (22) thus—

$$\text{h.p.} = \frac{pav}{33000} \quad . \quad . \quad . \quad . \quad (26)$$

which is often expressed as $$\text{h.p.} = \frac{u}{33000} \quad . \quad . \quad . \quad . \quad (27)$$

Furthermore, if P is the total pressure on the area *i.e.* $P = ap$, then (26) becomes

$$\text{h.p.} = \frac{Pv}{33000} \quad . \quad . \quad . \quad . \quad (28)$$

It will be seen that this series can be subjected to many further algebraic modifications.

The indicated horse-power, i.h.p., of a steam-engine is obtained by deriving the total ft.-lbs. developed from the product of the mean steam-pressure throughout the stroke in lbs. per sq. in. P, the length of the stroke, L, in feet, the area, A, of the cylinder in sq. in., and the number, N, of effective strokes per minute, then proceeding as above for h.p., thus—

$$\frac{P \times L \times A \times N}{33000} \quad . \quad . \quad . \quad . \quad (29)$$

The brake horse-power, b.h.p., is obtained by a dynamometer, which is a device for registering the maximum effort the engine is capable of in ft.-lbs. of useful work, *i.e.* the possible work procurable from the crank-shaft, and i.h.p. – b.h.p. = work lost in friction (in bearings, glands, pipes, etc.).

$$\therefore \text{ Efficiency} = \frac{\text{Useful work done}}{\text{Work transmitted to machine}} = \frac{\text{b.h.p.}}{\text{i.h.p.}} \quad (30)$$

Momentum is often defined as the amount of motion in a body; that is, it is possessed by a body in motion. It is proportional to the mass, m, of the body and its velocity, v, thus—

$$\text{Momentum} = mv \text{ or } \frac{Wv}{g} \text{ ft.-lb.-seconds} \quad . \quad . \quad . \quad . \quad (31)$$

By (7) we know that a force P acting on a mass m at rest

accelerates it, and $P = \frac{Wa}{g}$ where a is the acceleration. If P acts for a time t seconds we find by (7) that $P = \frac{Wv}{gt}$ or $\frac{mv}{t}$, but mv is the momentum, so that $\frac{mv}{t}$ is that acquired by the body each second. Then calling momentum M ($= mv$) we see that M is dependent on the velocity, and, therefore, must have a direction like it (see Newton's Second Law). So also change of M must depend on the change of velocity, or, in other words, on the acceleration a ft. per second^2, hence

$$\text{rate of change of momentum} = \frac{W}{g}a \quad . \quad . \quad . \quad . \quad (32)$$

The direction of the force is that of the acceleration and is proportional to it, *e.g.* if a body of mass $\frac{W}{g}$ has a velocity of v_1 at one instant and of v_2 after any later time t, due to a constant force, the rate of change of momentum is

$$\frac{\frac{W}{g}v_2 - \frac{W}{g}v_1}{t} \text{ or } \frac{W}{g}\frac{(v_2 - v_1)}{t} \quad . \quad . \quad . \quad (33)$$

the bracketed term $\left(\frac{v_2 - v_1}{t}\right)$ being the average acceleration which is constant for all values of t since the force is constant. Wherefore the acceleration is uniform and the rate of change of momentum is proportional to $\frac{W}{g}$ times a, as stated above in (32). This is exemplified in the case of falling bodies.

Impulsive forces are those acting for very short periods of time, the force P producing M being very great compared with the time t.

$$Pt = \frac{W}{g}(v_2 - v_1) = \text{impulsive force} \quad . \quad . \quad (34)$$

$$\therefore P = \frac{W}{g}\left(\frac{v_2 - v_1}{t}\right) = \frac{W}{g}a.$$

If the momentum acts at a distance r, the moment of momentum is $\frac{W}{g}vr$, and it will be seen that the change of the moment of the momentum is equal to the moment of the

impulse of the forces causing the change, *i.e.* to their angular impulse (see Newton's Third Law). This is usually stated thus: *change of momentum is proportional to the impulse of the impressed force.* If we consider the case of a fan, figs. 7 and 8, exhausting air from its centre outwards through straight radial blades, and having a peripheral velocity of V ft. per second, the air, which has to turn from an axial direction due to entry to a radial direction for exit, has a zero tangential velocity when about to begin its radial motion. The change of tangential velocity of the air being ejected is, then, the tangential velocity of the tips of the blades, and the change of momentum is mV if friction is disregarded. If the weight of

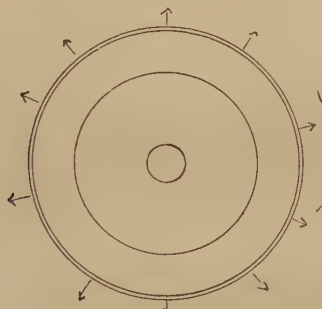

FIG. 7.

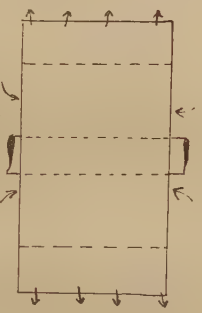

FIG. 8.

air dealt with by the fan is W lbs. per second, then the force exerted by the air is $\frac{WV}{g}$ lbs., which force the fan has to produce to effect ventilation. This force can be considered as acting at the circumference, and as the work done is the product of the force and velocity,

$$\therefore \text{ the work done per second} = \frac{WV}{g} \times V \text{ or } \frac{WV^2}{g} \text{ ft.-lbs.} \quad . \quad (35)$$

and the h.p. needed per second is

$$\text{h.p.} = \frac{WV^2}{g\,550} \text{ ft.-lbs.} \quad . \quad . \quad . \quad (36)$$

E.g. 100,000 cub. ft. of air per minute, weighing 0·08 lb. per cub. ft., is exhausted from a mine by a fan 20 ft. in diameter and running at 100 r.p.m. What is the h.p. needed to create flow of the air ?

By (36) $$P = \frac{WV^2}{g}$$

$$V = \frac{100 \times 20 \times \pi}{60} = \frac{2200}{21} \text{ ft. per second.}$$

$$\frac{W}{g} = \frac{100000 \times 0{\cdot}08}{32{\cdot}2 \times 60} = \frac{200}{48{\cdot}3}.$$

$$\frac{WV^2}{g\,550} = \frac{200}{48{\cdot}3} \times \frac{2200}{21} \times \frac{2200}{21} \times \frac{1}{550} = 82{\cdot}6 \text{ h.p.}$$

Friction is the force tending to prevent motion between surfaces in contact, and differs greatly in its nature according to the kinds of materials in contact and the physical state of the materials. Hence we are compelled to consider friction separately, according to whether the materials are in the solid, liquid, or gaseous states.

Solid friction, or that existing between solid surfaces, like liquid friction, enters very little into ventilation problems, but an understanding of the features exhibited in these cases will aid very materially in deducing the laws of mine friction later. The laws of solid friction are deduced from experimental observations of a mass caused to move over given surfaces by means of a weight attached to it by a string passing over a fixed pulley. It will be seen that the roughness of the surfaces in contact greatly affects the limiting force or friction to be overcome in the case of solids.

1. The limiting force, F, per unit weight, is independent of the area of the surfaces in contact. This can be proved by using a mass or block whose sides are of different area and testing for F by using each side in turn as the contact area. The weight being the same in each case, it will be found that F is also the same for any of the surfaces in contact. This result often appears to be contrary to fact.

2. The limiting value of F is directly proportional to the normal (perpendicular) pressure per unit area between the contact surfaces. This is shown by plotting the results of the friction F, encountered in overcoming the resistance R, by increasing the weight on the block or area without of course changing the area for each determination. The ratio $\frac{F}{R}$ will be constant.

The ratio $\frac{F}{R}$ is called the " coefficient of friction " of the given surfaces in contact, and is sometimes, but not wisely, stated as

that fraction of a body's weight needed to move it under the conditions existing.

The friction between two bodies at rest, or the statical friction, is slightly greater than that existing between them when moving relatively to each other, or dynamical friction.

3. The limiting value of F depends on the roughness and material of which the surfaces are made.

4. It is independent of the speed of sliding. Sliding friction is greater than rolling friction.

Liquid friction laws (*i.e.* the laws relating to friction between liquids and solids) are not nearly so well known or definable as the solid friction laws, but are approximately as follows :—

1. Liquid friction is proportional to the extent of the wetted surfaces.

2. It is independent of the boundary material, and, to some extent, of the roughness of that material.

3. It is dependent on the liquid pressure.

4. It varies inversely (but not regularly so) as the temperature of the liquid.

5. It increases with the speed of the liquid.

The behaviour of gases and air with regard to friction is more akin to that of liquid than that of solid friction, but gases are subject to so many variations by virtue of their three-dimensional properties that we must defer them at this stage to a later chapter, as must be done with several other definitions.

Heat.—The intermolecular forces of matter binding the whole system of molecules, of which any portion of matter may be said to be composed, can assert themselves, under different conditions, in various forms of energy. It is generally accepted now that the kinetic energy of the molecules of a body constitute the heat of the body. Hence we claim heat as a form of energy, and whenever work is done in overcoming friction heat is produced regularly—that is, for every foot-pound of work done a definite amount of heat is produced. Therefore the greater the amount of work spent on a body the greater the amount of heat produced.

Heat is transferred in three ways, viz.: (*a*) Conduction; (*b*) Convection; (*c*) Radiation.

(*a*) **Conduction** is the process whereby the heat is conveyed from particle to particle by molecular vibration until, finally, the most remote parts of the body are heated, as in the case of a copper wire one end of which is put into the fire; the other end soon becomes too hot to hold in the hand.

Metals, as a rule, are good conductors, *i.e.* they transmit heat rapidly; but minerals, especially coal, are poor conductors. Gases and liquids are also poor conductors.

Heat Conductivity (k).—This is the quantity of heat which will pass through a slab of material of unit thickness and unit cross-section in unit time when the temperature difference between the faces is 1°. Thus if s is the sp. ht. of the material, and d its density, the heat required to raise the temperature of unit volume through one degree is sd. The quantity of heat (k) which would pass through the above unit slab would raise the temperature of the material by t°.

$$\therefore \text{ the conductivity} = k = \frac{sd}{t}, \text{ whence } t = \frac{k}{sd} \quad . \quad (37)$$

Heat Diffusivity (K).—This is the rate of conducting heat when, as above, the unit of heat is that amount which will raise unit volume of the material 1° C. in temperature. It is obviously the second quotient above—

$$K = \frac{k}{sd} \quad . \quad . \quad . \quad . \quad (38)$$

(See Table XI., facing p. 94.)

(*b*) **Convection** applies to liquids and gases, but not to solids. It is due to the establishment of currents by the heat expanding, and therefore lowering the density of the fluid in its vicinity. This causes the fluid to rise, and, in so doing, it is displaced by the heavier portions of the same body until, ultimately, a circuitous current or flow is set up. Convection takes place above the source of heat, and is well illustrated in the case of a water-tube boiler or in furnace ventilation where the flow, as in all cases, is above the fire, which in this latter case is situated at the upcast pit bottom.

(*c*) **Radiation** is the term given to the transference of heat through an intervening space, and since such can happen even though the space be a vacuum it cannot be explained by either conduction or convection. It is known to be a wave or vibratory transference in the ether which permeates space, and indeed of which matter itself is said to be a form.

Temperature.—The number which expresses on some definite scale how hot a body is, is called its temperature. It is measured by means of thermometers and pyrometers, the latter being used only for high temperatures like those of furnaces, hearths, etc.

As bodies usually expand on heating and contract on cooling, and the principle of the thermometer depends upon this property of matter, if we adopt a body whose rate of expansion and contraction due to changes of temperature is very uniform, a suitable means will then be obtained for the measurement of temperature. Such a body is mercury ; a mercurial thermometer

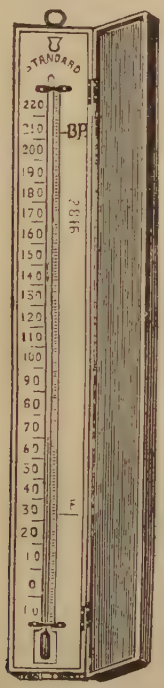

Fig. 9.—
Fahrenheit.

Fig. 10.—
Centigrade.

Thermometers.

is simply a glass tube of fine uniform bore, blown at one end and filled from the other or open end. When the tube is full it is heated to the highest temperature we desire to measure ; the open end is then hermetically sealed. The mercury is then allowed to contract, filling the bulb and a portion of the tube, leaving a vacuum above it to the sealed end. The temperature of melting ice, or freezing-point, and that of boiling water-vapour at a pressure of 760 mm., or boiling-point, are now marked on the tube for comparative measurements. These points are fixed

experimentally by immersing the bulb in melting ice and then in boiling water-vapour, and the stationary position of the mercury in each case etched on the glass. The space between these two points is divided up equally for "degrees."

There are three scales in use—

1. The Centigrade, C., with 0° freezing-point and 100° boiling-point.

2. The Fahrenheit, F., with 32° freezing-point and 212° boiling-point.

3. The Reaumur, R., with 0° freezing-point and 80° boiling-point.

It will be seen that the respective sensible temperature ranges (*i.e.* from freezing- to boiling-points) above are in the ratio—

$$100 : 180 : 80$$

or

$$5 : 9 : 4$$

so that the scales are interchangeable and can be easily converted from one to the other.

E.g. 1. **To change degrees C. into degrees F.,** multiply the C. degrees by $\frac{9}{5}$ and add 32°. The 32° is added because in the Fahrenheit scale, freezing-point is 32° F. above zero.

$$i.e.\ \text{F.}^\circ = \left(\frac{9}{5} \times \text{C}^\circ\right) + 32 \quad . \quad . \quad . \quad (39)$$

2. **To change degrees F. into degrees C.,** first subtract 32° from the F. reading because of the high reading for freezing-point F., then multiply by $\frac{5}{9}$.

$$i.e.\ \text{C.}^\circ = (\text{F.} - 32) \times \frac{5}{9} \quad . \quad . \quad . \quad (40)$$

The Reaumur Scale is not used outside of Russia and the Scandinavian countries, so that we need not discuss it here, though the same procedure holds, and is only modified according to the given scale ratio, *i.e.* 4 : 5 : 9.

Absolute Temperature is another means of measuring temperature and depends upon the expansive properties of gases, and will be treated later.

The unit of heat is the unit by which is obtained the measure of the amount of heat of a body, and it is distinguished from temperature in that the latter measures how hot a body is. The unit varies in different countries, *e.g.*—

The *calorie* is the amount of heat required to raise the temperature of 1 gramme of water 1° C. when at its maximum density (*i.e.* at 4° C. or about 39° F.).

The *large calorie* is the amount of heat required to raise the temperature of 1000 grammes of water 1° C. when at its maximum density. In addition to the above there is an *English calorie*, which is the amount of heat required to raise the temperature of 1 lb. of water 1° C. when at its maximum density.

British engineers express the unit of heat as the amount of heat needed to raise the temperature of 1 lb. of water through 1° F. when at its maximum density. This is called the *British Thermal Unit*, or *B.Th.U.*

1000 grams = 2·2 lbs. approximately
and 1° C. = 1·8° F.
∴ 1 large calorie = 2·2 × 1·8 = 3·96 B.Th.U.
or 1 B.Th.U. = 0·252 large calories or 252 calories.

Dr Joule found the relation between heat developed, as a result of work, and the amount of work done to be 1 B.Th.U. = 778 ft.-lbs. of work. This is known as *Joule's Mechanical Equivalent of Heat.* Different authorities give different values from 773 to 779.

1 B.Th.U. = 778 ft.-lbs. of work.
1 large calorie = 778 × 3·96 = 3081 ft.-lbs. of work.
1 calorie = 3·081 ,, of ,,

$$1 \text{ lb.–degree C. calorie} = 778 \times \frac{9}{5} = 1400 \text{ ft.-lb. of work.}$$

Coal when burnt can develop from 9000 to 15,500 B.Th.U. per lb. The average Welsh coal contains 14,000 B.T.U. per lb., that is,

14000 × 778 = 10,892,000 ft.-lbs. of energy per lb.

which gives a slight notion of the immense amount of energy stored in coal.

Specific heat (sp. ht.) is the ratio of the amount of heat required by a substance at 0° C. to raise its temperature 1° as compared with that required by an equal weight of water which is taken as unity.

e.g. Specific heat of water = 1·000.
,, ,, copper = 0·0936.
,, ,, iron = 0·089.
,, ,, coal = 0·240.

The sp. ht. of a substance varies with the temperature, and

with gaseous substances the change is very pronounced, especially if either the pressure or volume be kept constant.

TABLE I.—*Specific Heats of some well-known Gaseous Bodies found in Mines.*

Name of substance.	Chemical symbol.	Sp. ht. at constant pressure. S_p.	Sp. ht. at constant volume. S_v.	Remarks.
Dry air	..	0·2417	0·1715	20° C. temperature
Normal air	..	0·2389	0·169	" " "
Carbon dioxide	CO_2	0·202	0·1650	" " "
Carbon monoxide	CO	0·242	0·171	" " "
Nitrogen	N_2	0·235	0·175	" " "
Methane	CH	0·591	0 452	" " "
Sulphuretted Hydrogen	H_2S	0·245	0·1828	" " "
Water vapour	H_2O	0·4652	0·340	100° " "

When the sp. ht. of a body is stated without qualification, that at constant pressure is usually meant.

Latent Heat.—In order that matter may undergo a change of state, *i.e.* from solid to liquid or liquid to gas, heat must be supplied. During the change of state there is no temperature change, and consequently when ice is being melted into water the thermometer gives no indication of the heat supplied until after all the ice is converted into water. This hidden heat is termed the latent heat, and for the above case of the conversion of solid to liquid is known as the latent heat of fusion. Regnault, Bunsen, and others have shown that the latent heat of fusion of ice is about 80 calories per gram, or, in engineers' units, 144 B.Th.U. per lb.

The heat which in a similar manner is latent in the case of the conversion of a liquid to a gas is known as the latent heat of vaporisation of the substance being considered. A gram of water at 100° C. requires 537 calories to convert it into steam at 100° C., which heat is given out again on condensation to water at that temperature. The latent heat of vaporisation of water is given by Regnault as $L_t = 606{\cdot}5 - 0{\cdot}695t$ where L_t is the latent heat in calories per gram and t the temperature centigrade. This amounts to 537 cals. per gram, or 966 B.Th.U. per lb. at atmospheric pressure. The latent heat is

not constant but varies with the temperature and is given by Regnault for any temperature, t, Fahrenheit, by the formula $L_t = 966 - 0{\cdot}7\ (t - 212)$, B.Th.U. per lb.

Critical Data.—The critical temperature is the highest temperature at which a gas can be liquefied by subjecting it to pressure.

The critical pressure (of a gas or liquid) is the pressure at the critical temperature.

The critical volume is usually taken as the ratio of the volume of the gas at the critical temperature and pressure to that at N.T.P., which latter is usually considered unit volume.

The Heat of Combustion is the heat in calories given out when the molecular weight in grams of a substance is completely burnt at constant pressure.

The Heat of Formation is the heat liberated when the molecular weight in grams of a compound is formed from its elements. If in such case heat is absorbed, a minus sign is placed before the amount so absorbed. For organic compounds the heat of formation is deduced by subtracting the heat of combustion from the heat generated by the burning of the carbon and hydrogen contained in the compound.

Fig. 11.—Principle of the barometer.

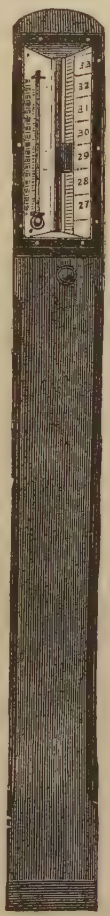

Fig. 12.—Colliery barometer (Davis).

Atmospheric Pressure.—The atmosphere forms a spherical layer about the earth some 200 to 300 miles high, attenuating upwards. We are thus subjected to the pressure of this sea of atmosphere. This pressure is called atmospheric pressure, and is measured instrumentally by the force it exerts against a vacuum in two ways, viz.—

(*a*) by the Mercurial barometer.

(*b*) by the Aneroid barometer.

(*a*) The mercurial barometer may be constructed by taking a glass tube, about a yard long, of uniform bore, and closed at one end only. The tube on being filled with clean dry mercury is exhausted of air. Now seal the open end with the thumb and insert the tube into a small cistern of mercury. Remove the thumb while it is below the level of the mercury in the cistern and the level of that in the tube falls to a height, *h*, between 27 in. and 31 in., dependent on the prevailing atmospheric pressure. It will be seen by fig. 11 that the pressure of the atmosphere downward upon the mercury in the cistern supports the column in the tube, which column will rise and fall with any fluctuation of the pressure. The space above the surface of the mercury in the tube is a vacuum, or nearly so, and is, like the experiment itself, named after its discoverer, Torricelli. The tube is conveniently graduated with a vernier scale (see fig. 12). The height of this column at sea-level is normally 760 mm., or 29·922 in., or, roughly, 30 in., and, since the weight of a cubic inch of mercury is 0·49 lb., then the normal atmospheric pressure is about $0{\cdot}49 \times 30 = 14{\cdot}7$ lbs. per sq. in.

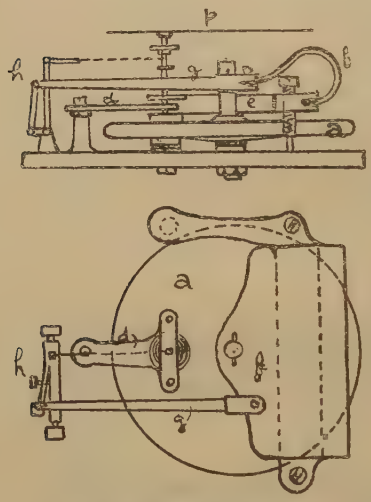

FIG. 13.—Aneroid barometer. (Internal mechanism.)

The mercurial dial, fig. 14, in general use is actuated by the movement of a glass float, *f*, in the turned-up open limb, L, of the mercury column, attached to a small glass counterpoise, *g*, by a thread, *t*, passing over a wheel, *w*. Any rise or fall of the mercury causes the wheel to rotate by moving *f*, and in so doing the pointer P, through which is soldered the spindle, revolves also. A conveniently graduated circle, C, shows the particular atmospheric pressure, so enabling anyone to "read the barometer."

(*b*) Aneroid barometers depend in principle upon the sensitive responses, to fluctuations of atmospheric pressure, of a sealed box from which the air has been exhausted. This box, *a*, fig. 13, is often corrugated to increase the pressure effect. It expands

with fall and contracts with increase of pressure. In so doing the mechanism, which consists of levers d, e, f, g, and h, directly

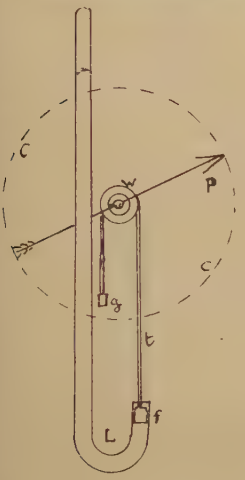

Fig. 14.

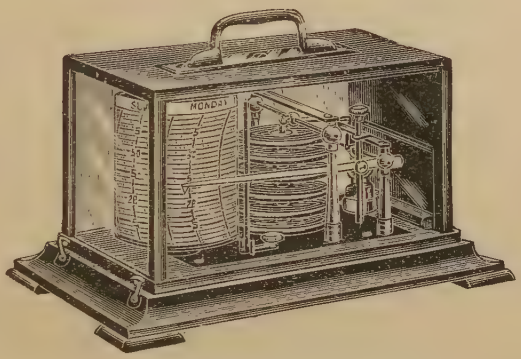

Fig. 15.—Barograph, or self-recording aneroid barometer (Thornton).

attached to the box, actuates a pointer, p, working over a graduated circle.

It will be readily noted that such a simple contrivance is easily

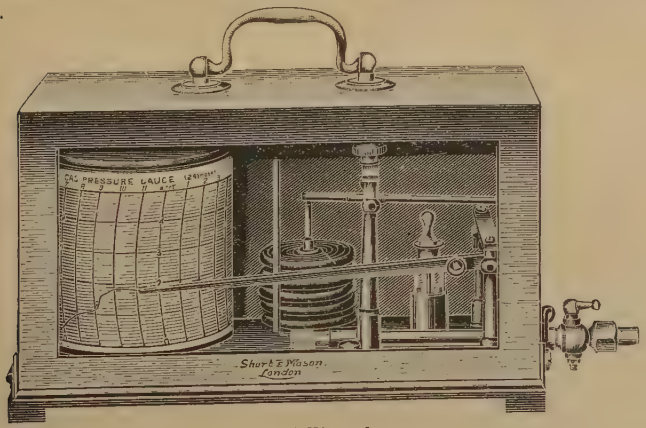

Fig. 16.—Colliery fan gauge.

adapted to a clockwork-operated graph-drum, so giving a daily or weekly graph or barograph. Fig. 14 shows the mechanism

of the dial-faced mercurial barometer. Fig. 15 shows an aneroid barograph.

We have seen that pressure varies inversely with the altitude. This fact is taken advantage of in the case of barometers for surveying purposes. Roughly, 900 ft. of altitude, or depth,

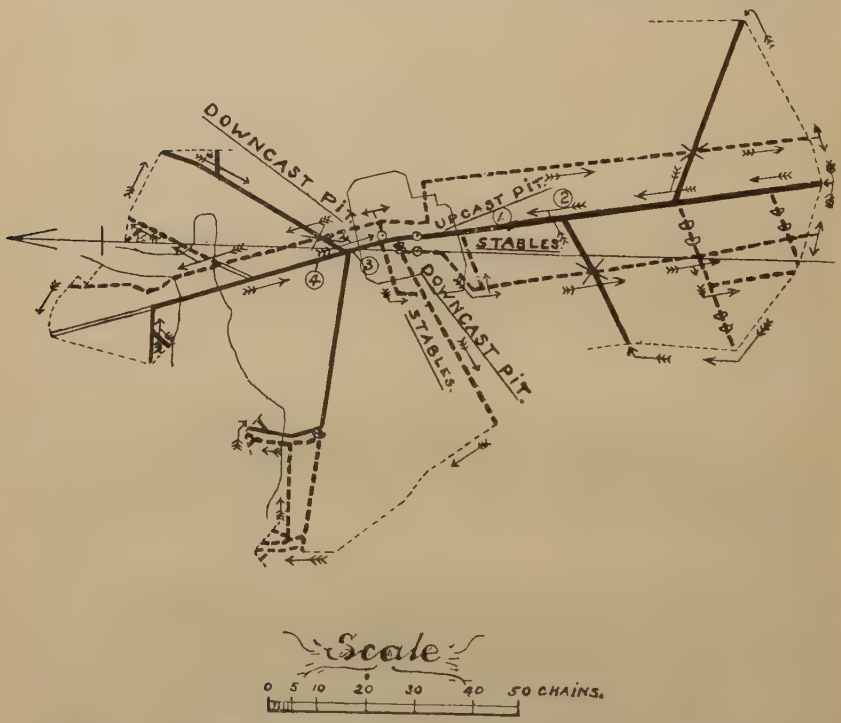

FIG. 17.—From *Notes on the Analyses of Mine Air*. xxviii. 2. S.W.I.E. By J. W. Hutchinson and Edgar C. Evans. (By the courtesy of the South Wales Institute of Engineers.)

equal 1 in. of barometer reading, and, for very general purposes, we could construct a formula on the above principles which must necessarily be only approximate, since the atmosphere is not homogeneous, but thins out quickly as we leave the earth.

Air at ordinary T. and P. weighs 0·0766 lb. per cub. ft. approx.

Water weighs 62·3 lbs. per cub. ft. approx.

Mercury ,, 13·6 times as much as water.

TABLE II.—CORRESPONDING WITH CHART, FIG. 18.

Results of Analyses at places (1) *and* (2) *of Colliery*, fig. 17.

Date.	Percentage of methane.		Barometer in inches.
	Place (1).	Place (2).	
January 8	1·02	1·52	29·0
„ 9	1·35	1·66	29·4
„ 10	1·38	1·70	29·8
„ 11	1·21	1·77	29·87
„ 12	1·24	1·57	30·09
„ 13	1·36	1·80	29·90
„ 14	..	..	..
„ 15	1·34	1·75	29·63
„ 16	1·45	1·72	29·47
„ 17	1·32	1·62	29·71
„ 18	1·32	1·56	29·70
„ 19	1·20	1·59	29·90
„ 20	1·42	1·83	29·73

TABLE III.—CORRESPONDING WITH CHART, FIG. 19.

Results of Analyses at places (3) *and* (4) *of Colliery*, fig. 17.

Date.	Percentage of methane.		Barometer in inches.
	Place (3).	Place (4)	
January 8	1·20	..	29·00
„ 9	1·31	..	29·40
„ 10	1·28	1·42	29·80
„ 11	1·26	1·27	29·87
„ 12	1·16	1·24	30·09
„ 13	1·37	1·57	29·90
„ 14	..	..	..
„ 15	1·45	1·62	29·63
„ 16	1·53	1·60	29·47
„ 17	1·32	1·50	29·71
„ 18	1·24	1·45	29·70
„ 19	1·31	1·35	29·90
„ 20	1·46	1·71	29·73

$\therefore$ Mercury is $13{\cdot}6 \times \frac{62{\cdot}3}{0{\cdot}0766}$ times, say 11,000 times, as heavy as air.

i.e. 1 in. of Hg will support approximately 11,000 in. of air.

" " " " 900 ft. "

Taking a 30-in. column of mercury as the height due to the atmospheric pressure, 30 in. = 30 × 900 ft. = 27000 ft. of air in a *homogenous* column is needed to support this pressure. From this we can construct the following rough formula :—

$$I = \frac{DB}{27000} \quad . \quad . \quad . \quad . \quad (41)$$

where D = depth of shaft in feet.
B = barometer reading in inches.
I = change of barometer reading due to D.

The barometer is of extreme importance in mines, as by it we are able to learn what pressure the atmosphere is setting up to counteract the tendency of gases to exude from crevices, pores, etc., in the strata. If the barometer reading falls, the mine atmospheric pressure has fallen and noxious gases may be expected, for it will then be easier for them to accumulate, there being less atmospheric pressure to overcome. Study carefully Tables II. and III., and charts, figs. 18 and 19.

The C.M.A., 1911 (sec. 71), states that a barometer and thermometer shall be placed in a conspicuous place above ground near the entrance to the mine, and (by General Regulations) to be read by all responsible persons before entering the mine. These readings are to be entered in a book specially kept for the purpose.

Humidity is the term applying to the amount of moisture or water-vapour in the atmosphere. It is not constant, the quantity present depending upon the temperature. As compression evolves heat, so evaporation produces cold, therefore in warm weather and tropical countries the air is more humid than in cold weather or places. When air contains its maximum amount of vapour—which amount increases with the temperature—it is said to be *saturated*, but if further heated its capacity for water vapour is increased, whilst if it is cooled it deposits its vapour as liquid water. This point at which the air or any gas deposits its water vapour is called the *dew point*.

Air is seldom saturated, so it is convenient to express any humidity as a percentage of that needed at that time and tem-

perature to saturate the air. Thus we speak of *Relative Humidity*,

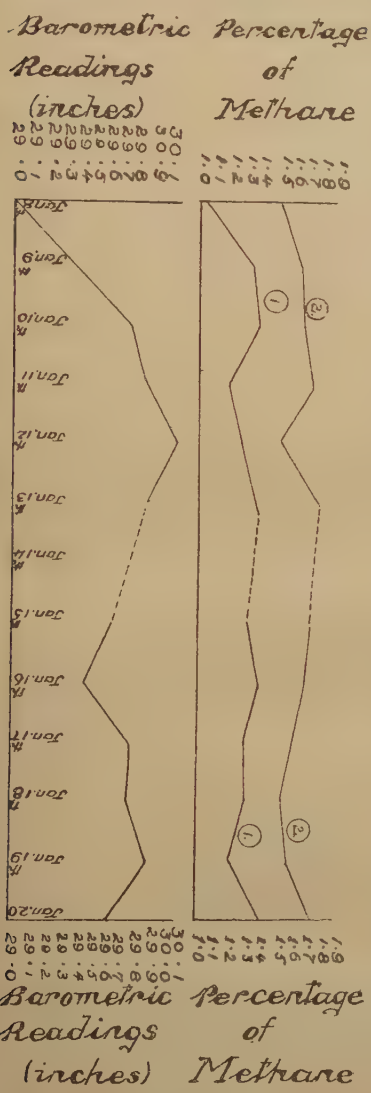

FIG. 18.

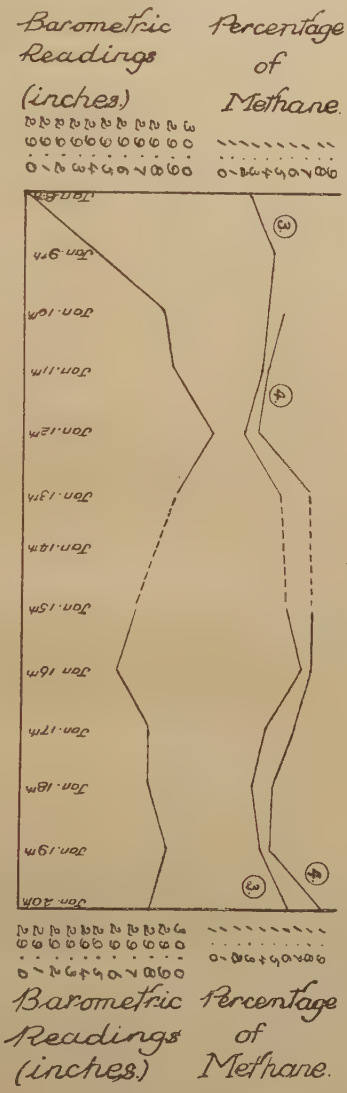

FIG. 19.

e.g. if the air at any time contains three-quarters of the amount of water vapour it is capable of holding, then there is a relative

humidity of 75 per cent., the saturation-point being 100 per cent. Winds or currents of fresh air assist the evaporation of the moisture in that they supply portions of fresh air to be saturated by the water vapour. Thence the importance of fresh air currents in zones of high relative humidity. The relative humidity is measured by the hygrometer (see fig. 20), in which A is the dry-bulb and B the wet-bulb, the latter having its bulb encased in muslin. This muslin cover ends in a skein leading to a small reservoir of water, C, up which water rises and on evaporation cools the bulb, so registering a lower temperature on the B than on the A scale. The dry-bulb measures the natural temperature. Now, if the water cannot evaporate upon the B bulb no heat is abstracted, and therefore there is no difference in the two readings; this can only occur when the air is saturated. On the other hand a difference between the readings shows that the air is unsaturated and its relative percentage is calculated from hygrometic tables (see Glaisher's Tables supplied at the end of the book).

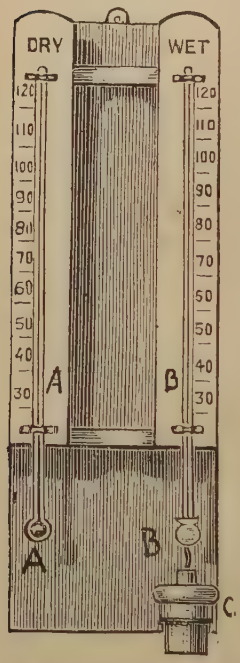

FIG. 20.—The hygrometer. (Mason's.)

Highly humid atmospheres create dangerous physiological conditions and lower the working temperature limit for, and physical capacity of, men. Very dry atmospheres assist the accumulation of dust; a happy mean is usually maintained or sought. The problems arising out of humidity will be treated in the chapter on "Mine Temperatures and Humidity." The Mines Act (C.M.A., 1911), sec. 71, requires that hygrometers be placed below ground in a conspicuous place near the shaft in the main intake airway and in the main return airway. By General Regulations it is required that a responsible official must read the intake airway hygrometer daily and the one in the return airway weekly, and enter the readings in a book to be kept at the mine for the purpose.

The Recording Hygrometer or Psychrometer.—Figs. 21 and 22 show the appearance and mechanism of the Bristol recording

psychrometer, or humidity recorder, the operation of which is based on the usual hygrometer principle that evaporation varies with the amount of moisture in the atmosphere. The temperature of the wet-bulb is always lower than the surrounding atmosphere. The difference in the temperature readings of the wet-bulb and dry-bulb is known as the wet-bulb depression. This varies directly with the rate of evaporation, which rate is inversely proportional to the amount of water vapour in the air, at the indicated atmospheric temperature. By taking the dry-bulb temperature and noting the wet-bulb depression, the relative humidity in per cent. can be readily ascertained from the tables at end of this book.

A flexible capillary tubing connects the sensitive bulbs and the instrument. This long-distance feature permits the instrument and bulb to be installed several feet apart. It is the temperature at the sensitive bulb which is recorded on the instrument. Any changes in temperature along the connecting tube do not affect the accuracy of the equipment. The bulbs are usually located at the required place, while the instrument is removed at a convenient place for observation and necessary attention. The standard length of tubing is 25 ft.

The bulbs are mounted on a water reservoir of the continuous-flow type. The reservoir is made of aluminium, therefore it will not rust. The design and construction permits the bulb to be supported in such a manner that the sensitive portions do not come in contact with any metal of the reservoir. Thus, when rapid fluctuations of temperature take place the readings recorded are in no way affected by the temperature of the metal reservoir. The dry-bulb is at one side, sufficiently distant to be unaffected by the evaporations. Details of the reservoir fixture are shown in line sketch (fig. 22).

There are two water reservoirs, upper and lower. The union marked "water supply," leading directly to the upper reservoir, should be connected with a permanent water supply. The flow of water is regulated by a valve in the line, and should be just sufficient to maintain a constant level in both upper and lower reservoirs. This makes it important that the reservoirs be installed in a level position. The union marked "overflow" leads from the lower reservoir and should be connected to a drainage system.

A wick fits over the wet-bulb and extends to both the upper and lower reservoirs. The flow of water causes a continuous

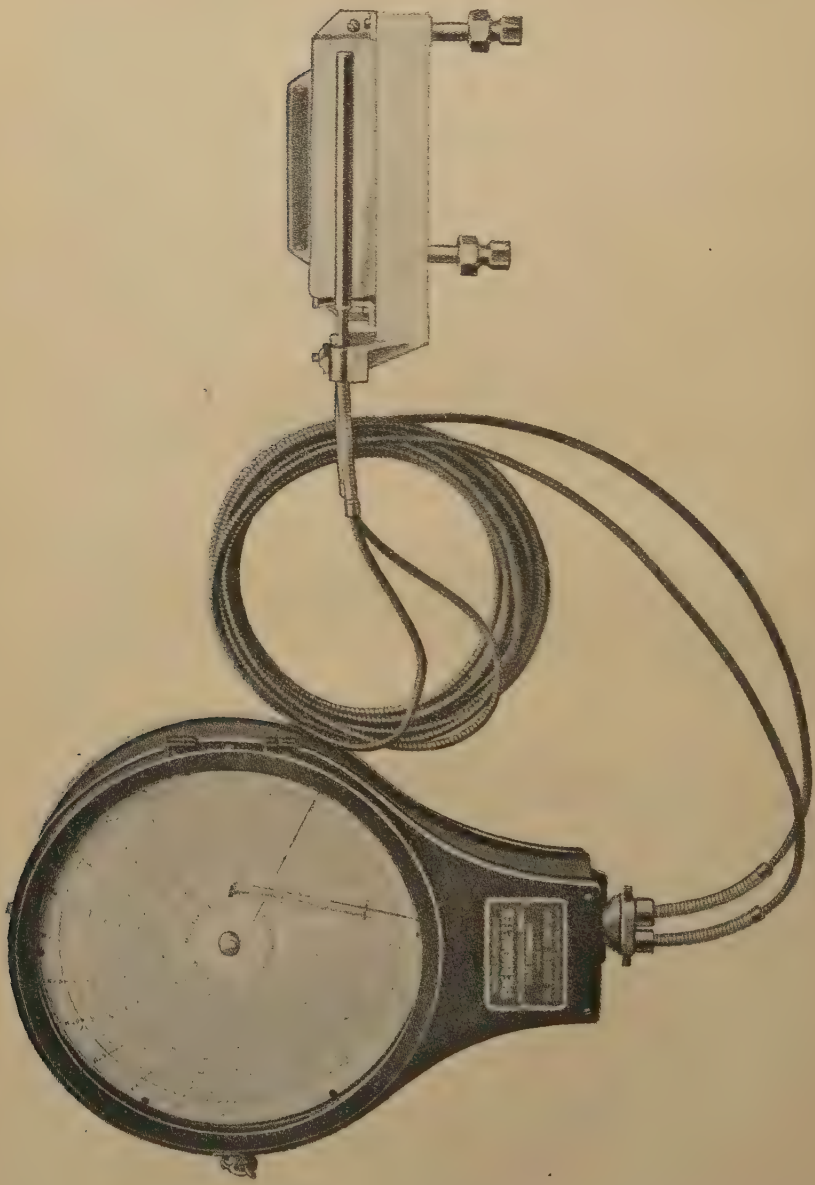

FIG. 21.—Recording psychrometer. (The Bristol Co., U.S.A.)

film to be maintained over the surface of the wick, and insures a permanently moist and clean wet-bulb.

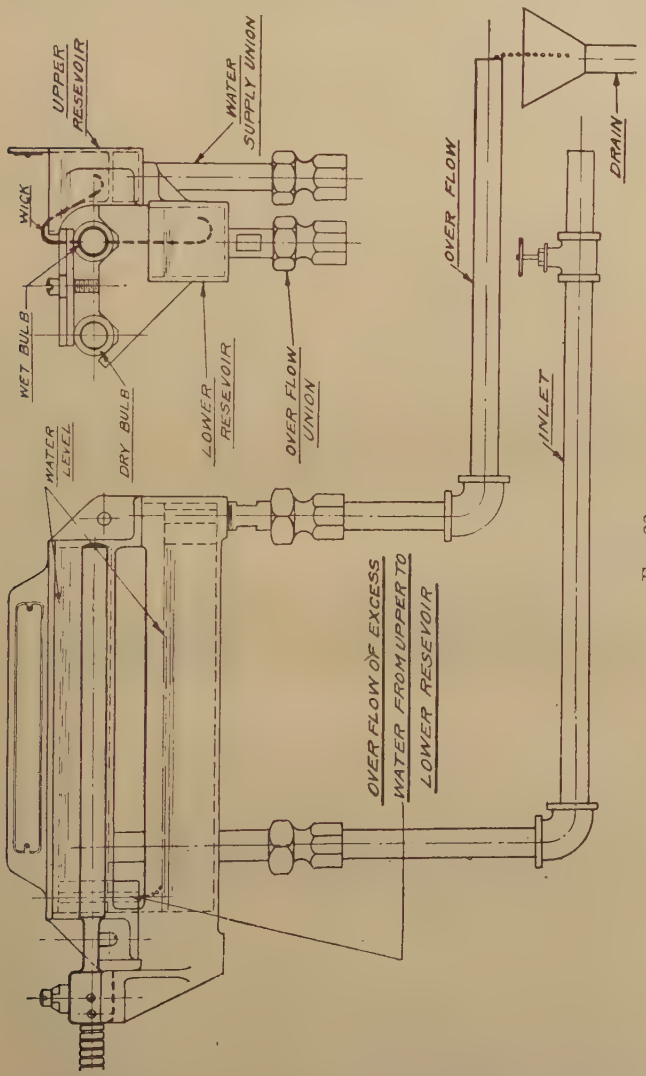

Fig. 22.

The Anemometer is an instrument used for the measurement of velocity. In addition to the above periodic instrumental checks required by the C.M.A., 1911, there will be found in

sec. 29 (2) laws demanding an entry in a special book at least once a month of the amount of air passing in the main air current and every split. By sec. 20 (2) the measuring points are to be shown on the plan, the intake air-ways coloured blue and returns coloured red.

These measurements are taken by means of the anemometer,

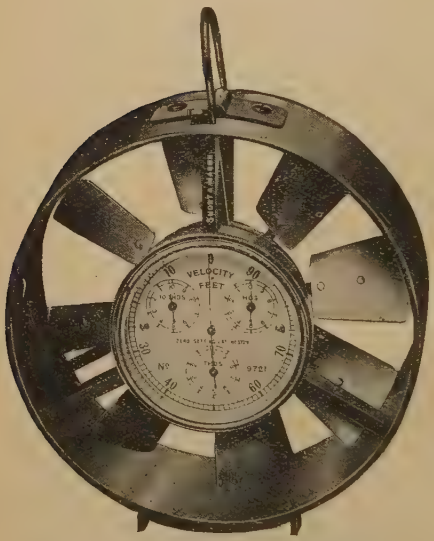

Fig. 23.—Biram anemometer (Short and Mason).

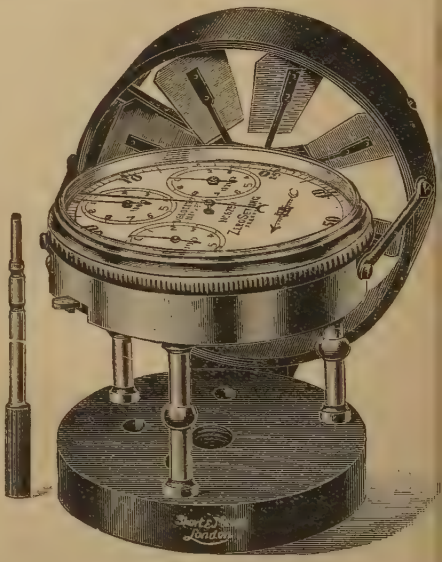

Fig. 24.—Short and Mason airmeter.

which gives the velocity; this, multiplied by the area, gives the quantity by formula (23). The anemometer can only be used for simple work like the above measurements of main velocities. It consists of a small wheel 3 in. to 6 in. diam. (fig. 23), in which vanes are set inclined to the plane of rotation, so that when a current passes axially through the wheel it will rotate. The motion of the wheel is communicated to gearing which works a counter, showing the number of revolutions in tens and multiples of ten up to a million feet by several dials in some types. It has a thumb lever clutch or cut-out to facilitate timing and calculations.

Usually the anemometer exaggerates. It is graduated by rotation in still air at the end of an arm of fixed radius, and for more accurate work should be recalibrated periodically as it

ages and a modification scale supplied. The instrument should be held on a rod away from the body in front of, but not in line with, the current of air. In taking readings of the velocity over any section of roadway the area should be split up into imaginary squares as in fig. 25, and the mean of the readings

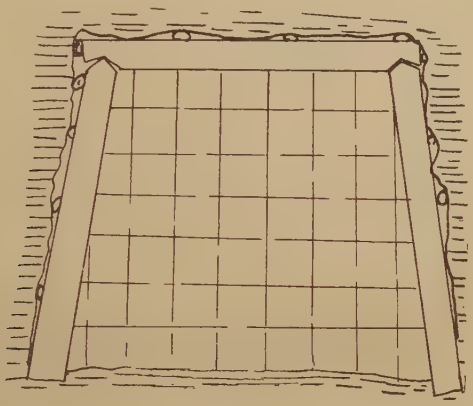

Fig. 25.

adopted, because the velocity near the sides is reduced as a result of surface friction. The practice of moving the instrument in a sinuous curve laterally from roof to floor is not recommended, because it adds the velocity due to its own movement to that of the air current.

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CHAPTER II.

FUNDAMENTAL LAWS AND THEOREMS.

Absolute Temperature.—The increase in volume at 0° C. or 0° F. which the atmosphere or any gas undergoes when heated, is the same for all gases. In the centigrade system of temperature measurement this increase is $\frac{1}{273}$rd of the volume *at* 0° *C.* for each degree C. rise in temperature. Now since $\left(273 \times \frac{9}{5}\right) - 32 = 459$ we may also say that all gases expand $\frac{1}{459}$th of their volume *at* 0° *F.* for each degree F. rise in temperature.

Therefore 459	cubic feet of gas	at	0° F.	
becomes 460	,,	,,	,,	1° F.
,, 461	,,	,,	,,	2° F.
,, $459 + t$	,,	,,	,,	t° F., and so on.

Since the fact also applies to "minus temperatures," it will be clear that were a gas to be cooled down to − 273° C. or − 459° F. its volume would theoretically vanish. Expansion is due to increased molecular activity set up by heat, and by decreasing the heat effect down to the above ultimate limit the molecular vibration would cease. This low temperature, − 273° C. or − 459° F., is known as the absolute zero, and any ordinary thermometer readings, if added to the particular one, C. or F., in which scale it is read, will give the absolute temperature usually denoted by T, *e.g.*

a temperature of t° C. = $(273 + t)$ ° C. absolute temperature.
or ,, ,, t° F. = $(459 + t)$ ° F. ,, ,,

Some authorities use 460, others 461, in dealing with the absolute temperature F., but for most purposes the difference is negligible.

Charles' or Dalton's Law.—This Law states that all gases

expand $\frac{1}{273}$rd of their volume at 0° C. (*i.e.* $\frac{1}{459}$th of their volume at 0° F.) for each degree C. (or F.) rise in temperature *provided the pressure is kept constant.*

Considering the fall of temperature we may proceed thus on the Fahrenheit Scale :—

F. scale temperature.	Absolute temperature.	Volume of gas.
−458° F.	= 1° T.	1
−100° F.	=359° T.	359
$-t$° F.	$=(459-t)$° T.	$459-t$.

which shows that the volume of a gas is proportional to its absolute temperature, so the above law may be stated thus: *When the pressure is constant the volume of a gas is in proportion to its absolute temperature.* From this, if V_1 is the volume of a gas at any temperature t_1, and V_0 is its volume at 0° F., then

$$V_1 = V_0\left(\frac{459+t}{459}\right)$$

or

$$V_1 = V_0\left(1+\frac{t}{459}\right) \quad . \quad . \quad . \quad (42)$$

If, on the other hand, the volume V_1 of a gas at temperature t_1° F. and the volume V_2 at any other temperature t_2° F., is given or sought, as is general, then

$$\frac{V_2}{V_1} = \frac{(459+t_2)}{(459+t_1)} \quad . \quad . \quad . \quad . \quad (43)$$

E.g. if 100,000 cub. ft. of air per minute enters a mine at 61° F., and, on returning up the fan drift its temperature is 91° F., due to the many heating agents in the mine, what is the volume of air leaving the mine considering the pressure constant ?

by (43)
$$\frac{V_2}{V_1} = \frac{(459+t_2)}{(459+t_1)} = \frac{T_2}{T_1} = \frac{\text{abs. temp. of } t_2}{\text{abs. temp. of } t_1}.$$

or
$$V_2 = \frac{V_1 T_2}{T_1}$$

i.e.
$$V_2 = V_1\frac{(459+91)}{(459+61)} = 100000 \times \frac{550}{520}$$
$$= 105769{\cdot}26 \text{ cub. ft. of air leaving the mine per minute.}$$

In the preceding discussion only the effect of temperature on the volume of a gas was considered, but it will be clear that pressure differences also affect the volumes to an appreciable extent. The effect of pressure was first investigated and correctly formulated by Robert Boyle, the gist of whose discovery in this department of research is always stated in a law which bears his name.

Fig. 26.

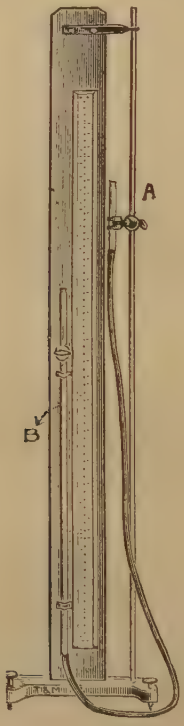

Fig. 27.

Boyle's Law.—The volume of a gas varies inversely with the absolute pressure when the temperature is constant.

This law can be easily demonstrated by means of a U-tube, as in fig. 26, closed at the lower turned-up end. When mercury is poured in by the open top end the air in the tube being unable to escape is compressed against the closed end. This space becomes measurably smaller at each addition of mercury, *i.e.* increase of pressure. A better way, more conducive to accuracy and convenience, is by means of the Boyle's Law Apparatus,

fig. 27, in which B is the closed tube fixed on the scale and A is the movable tube by raising or lowering which we may vary the pressure on the space in B. It will be found on tabulating or graphing pressures against volumes that the volume, v, of the air diminishes in exactly the same proportion as the pressure, p, increases.

Thus if p is multiplied by 2, the volume, v, is multiplied by $\frac{1}{2}$

" " " " 3, " " " $\frac{1}{3}$

" " " " 10, " " " $\frac{1}{10}$

" " " " x " " " $\frac{1}{x}$

from which in any of these instances v varies as $\frac{1}{p}$

or $v \propto \frac{1}{p}$

or $p \times v = k$ (a constant for the particular gas being treated) . . (44)

The effect of pressure being as important as that of temperature, the volume of a gas is not exactly defined by just stating its cubical capacity; we must also state at what temperature and pressure it occupies that volume. Hence the need for a standard; as stated in Chapter I., N.T.P. (760 mm. mercury and at a temperature of 0° C.) or the engineers' standard (30 in. mercury and 60° F.).

We have seen that the expression, $pv =$ constant, gives a fixed number for any given gas. If p_1 and v_1 are the respective pressure and volume of a gas or air at a certain instant, and p_2 and v_2 these quantities at another time, then if the temperature be taken as constant

$$p_1v_1 = p_2v_2 = \text{constant} \quad . \quad . \quad . \quad (45)$$

and if given any three of these we can find the fourth by proportion.

E.g. At 30 in. " reading of barometer " the volume of a given weight of air is 100,000 cub. ft., what volume will it occupy if the barometer " falls " 1 in., temperature remaining unaltered?

By (45) $p_1v_1 = p_2v_2$

$$\therefore v_2 = \frac{p_1v_1}{p_2}$$

$$\therefore v_2 = \frac{30 \times 100{,}000}{29} = 103448{\cdot}26 \text{ cub. ft.}$$

i.e. if pressure falls, volume rises.

Further consideration will convince one that to keep temperature constant, whilst varying the pressures and volumes, is in its entirety impossible, though one may approach very near to this condition of things. We could only achieve this equality of temperature under the above circumstances by utilising a perfect conducting material in our work or experiments, and also working very slowly. Any heat generated due to compression would then be completely conducted away as it was produced. Again, if the gas were allowed to expand, work is then being done *by* the gas and not *on* it as above, and the gas supplements this output of its energy by taking heat from its surroundings, so producing cold. One experiences this when a highly volatile oil is allowed to evaporate on the hand; a feeling of numbness, due to the cooling effect of evaporation, is felt. If, then, we could supply sufficient heat to the expanding gas, as the pressure is released, so as to maintain an equal temperature as in the above case of compression, we should attain equal temperature expansion just as we should have produced equal temperature compression. This condition of compression and expansion under equal temperature is called an **isothermal** condition, and for the lack of perfect materials is never entirely accomplished. On the other hand if we could, by means of perfect non-conducting materials, confine to a gas or air any heat set up due to pressure, or prevent any heat being absorbed by it during periods of release of pressure, *i.e.* expansion, the conditions would then be said to be **adiabatic.**

The adiabatic relation between pressure and volume is not $pv = k$ as in (44), but

$$pv^n = k \text{ (a constant)} \quad . \quad . \quad . \quad (46)$$

the n being the ratio (for air).

$$n = \frac{\text{sp. ht. of air at constant pressure}}{\text{sp. ht. of air at constant volume}} = \frac{0{\cdot}2417}{0{\cdot}1715} = 1{\cdot}408.$$

Now considering the laws of Boyle and Charles in conjunction we can denote the isothermal energy of p lbs. per sq. ft. of v cub. ft. of gas or air at an absolute temperature, T, thus—

$$pv = \text{TR ft.-lbs.} \quad . \quad . \quad . \quad . \quad (47)$$

In a similar manner we can express the adiabatic form given in (46); where R is the constant for air of 53·34, and the pressure, volume, and absolute temperature are stated in the terms as given above. This coefficient R varies inversely as the density

of the substance being considered. The following table gives a few values of R in the above terms deduced from Kaye and Laby's well-known work.*

(As in previous cases for illustration the figures are correct only to first place.)

TABLE IV.

Name.	Density. H=1.	R.	Remarks.
Air	14·4	53·34	*e.g.* R. for methane $= \frac{53{\cdot}34 \times 14{\cdot}4}{8}$ $= 96{\cdot}012$
Oxygen O_2	16·0	48·0	
Hydrogen H_2 . . .	1·0	776·0	
Carbon dioxide CO_2 . .	22·0	35·0	
Methane CH_4 . . .	8·0	96·0	
Carbon Monoxide CO . .	14	54·8	
Sulphuretted hydrogen H_2S .	17	45·0	

The constant R is a function of the density, and so varies with the temperature and pressure of the gas; it is conveniently obtained by expressing the difference between the specific heats at constant pressure and volume in units of work, *i.e.*

$$R = 778\ (S_p - S_v).$$

E.g. taking the figures for the specific heats for normal air as given in Table I., find the constant.

$$R = 778\ (0{\cdot}2389 - 0{\cdot}169) = 54{\cdot}38.$$

The difference in the case of carbon dioxide is due to the considerable deviation of this gas from the laws of Boyle and Charles; in an opposite sense the same remarks apply to hydrogen.

The Density of Gases and Air.—We have seen in Chapter I. that air is 14·4 times as heavy as hydrogen; but this implies a standard temperature and pressure, since the density of any volume of gas will increase under pressure and decrease under expansion or rise of temperature. It follows, then, from Charles' Law that *change of density due to pressure is a direct change, i.e.* for an increase or decrease of pressure there is a directly proportionate increase or decrease of density, provided the temperature remains

* *Physical and Chemical Constants*, by Kaye and Laby (Longmans).

constant. Thus, given a volume of gas of density d_1 at a pressure p_1 and of density d_2 at any other pressure p_2—

$$\frac{d_1}{d_2} = \frac{p_1}{p_2} \quad . \quad . \quad . \quad . \quad . \quad (48)$$

E.g. if, due to a barometric pressure of 29 in. of mercury, a cubic foot of air weighs 0·0766 lb., what will it weigh when the barometer indicates 30 in., if temperature be considered unchanged ?

From (48) by inversion—

$$\frac{d_2}{d_1} = \frac{p_2}{p_1}$$

$$\therefore d_2 = \frac{d_1 p_2}{p_1} = \frac{0{\cdot}0766 \times 30}{1 \times 29}$$

$$= 0{\cdot}0792 \text{ lb.}$$

Now considering pressure as being constant, we know from Charles' Law that rise of temperature expands or rarefies a gas, *i.e.* lessens its density, so we may state the law thus : *Change of density due to temperature is in the inverse ratio of the absolute temperature* provided pressure be constant.

Thus given a volume of gas of density d_1 and absolute temperature T_1, then, if the absolute temperature be T_2 and the density d_2 at any other instant, the above law may be written thus—

$$\frac{d_1}{d_2} = \frac{T_2}{T_1}$$

or

$$d_2 = d_1\left(\frac{T_1}{T_2}\right) \quad . \quad . \quad . \quad . \quad (49)$$

E.g. air weighing 0·0807 lb. per cub. ft. at 32° F. is heated up to 82° F. in traversing a mine, what will be the weight of a cubic foot leaving the mine ?

From (49) $$d_2 = \frac{d_1 T_1}{T_2} = \frac{d_1(459+32)}{(459+82)}$$

$$= 0{\cdot}0807 \times \frac{491}{541}$$

$$= 0{\cdot}073 \text{ lb. per cubic foot.}$$

Just as we summarised certain laws in (47), so we may combine

(48) and (49) and find the dual effect of the quantities treated in these cases thus—

$$(a)\quad d_1 : d_2 = p_1 : p_2$$
$$(b)\quad d_1 : d_2 = T_2 : T_1$$

Combining (a) and (b) $d_2 = \dfrac{d_1(T_1 p_2)}{(T_2 p_1)}$ . . . . (50)

E.g. under a pressure of 30 in. of mercury and at 32° F. a cubic foot of air weighs 0·08 lb. What is the effect on the density if the barometer "falls" to 28·5 in. and the temperature increases to 81° F. ? (Here the pressure is lowered and temperature increased, both of which lessen the density.)

By (44) $$0{\cdot}08 \times \left(\frac{459+32}{459+81}\right) \times \frac{28{\cdot}5}{30}$$
$$= 0{\cdot}0691 \text{ lb. per cubic foot.}$$

A vapour may be deemed a gas almost in its liquid state.
A gas ,, ,, vapour advanced from its liquid condition.

The Weight of a Cubic Foot of Air.—We have hitherto calculated the weight of a cubic foot of air only by comparing the effects of temperature and pressure on an already given weight per cubic foot. To calculate it from first principles requires an inquiry into the properties of aqueous vapours in addition to the weight of the air itself. The variation in the saturation pressure and volume of water vapour due to temperature, though thoroughly investigated by Regnault, cannot be expressed by any simple law. We are therefore obliged either to use tables, or, as is more general, to commence our discussion by accepting the weight of a given volume of air under stated conditions, as found by some reliable authority. Thus Atkinson, after many experiments, concluded that 459 cub. ft. of air at 0° F., and under a pressure of 30 in. of mercury, weighs 39·76 lbs., therefore—

if the pressure be 1 in. of Hg. the weight is $\dfrac{39{\cdot}76}{30}$ or 1·3253 lbs.
,, ,, ,, 2 ,, ,, ,, ,, 2·6506 lbs.
,, ,, ,, B ,, ,, ,, ,, $1{\cdot}3253 \times B$ lbs.

The weight of 1 cub. ft. at 0° F. would be $\dfrac{1}{459}$th of these figures, and for any barometric pressure, B, and temperature, t° F., by

the thermometer, the weight, w, of 1 cub. ft. of air may be stated as

$$w = \frac{1{\cdot}3253 \times B}{459 + t} \text{ lbs.} \quad . \quad . \quad . \quad (51)$$

The late Professor O'Shea found a slight error of 0·2 per cent. in the above constant by adopting the generally accepted weight of a cubic foot of air at 32° F., and under a pressure of 30 in. of mercury, *i.e.* 0·0807 lb. He thus obtained a constant of 1·32079. The U.S.A. Coast and Geodetic Survey, after very elaborate precautions and measurements, give the weight as 0·080728 lb. at 32° F. at sea-level and 29·92 in. of mercury (the correct mean) which when checked to zero F. gives 1·3212 as the constant, so the above formula becomes

$$w = \frac{1{\cdot}3212 \times B}{459 + t} \quad . \quad . \quad . \quad . \quad (52)$$

We shall adopt this figure as being between the two stated authorities and also because it gives an accurate check on the formula for the weight of a cubic foot of dry air as given in (53) or (54) *when used in conjunction with the formula for aqueous vapour pressure as in* (80) *or from Table* VI.

Sir R. A. S. Redmayne * gives the weight, D, of a cubic foot of *dry* air as

$$D = \frac{1{\cdot}3258 \times B}{459 + t} \text{ lbs.} \quad . \quad . \quad . \quad (53)$$

This weight of a cubic foot of dry air can also be deduced from first principles by considering the absolute temperature, T, the normal density of water, 62·3 lbs., the standard atmospheric pressure, 29·92 in., of mercury, and noting that mercury is 13·6 times as heavy as water; thus, by (47)

$$D = \frac{29{\cdot}92 \times 13{\cdot}6 \times 62{\cdot}3}{12 \times 53{\cdot}34 \times T} = \frac{39{\cdot}8}{T} = \frac{2{\cdot}707 \times \text{absolute pressure}}{\text{absolute temperature}} \quad (54)$$

This expression is more conveniently stated by taking, P, the corrected dry atmospheric pressure in lbs. per square inch and the actual temperature, t, thus—

$$D = \frac{P \times 144}{(459 + t) \times 53{\cdot}34} \quad . \quad . \quad . \quad (56)$$

Here 53·34 is the R of (47), *i.e.* the constant for air.

The vapour or saturation pressure due to moisture in the air

* Vol. iv., *Modern Mining Practice.*

depends on the temperature, and is proportional to it. This can be shown by introducing into the Torricellian vacuum of a barometer a drop of water by means of a pipette dipping under the mercury surface in the cistern up into the column. The water will rise to the surface of the column. Then if the column be steam or hot water jacketed, with a suitably connected glass tube, the temperature of the drop of water will rise, causing its water vapour to depress the mercury column until at the boiling of the water the column of mercury has sunk to the level of the mercury in the cistern. This demonstrates the tension of aqueous vapour (see fig. 28), which will be discussed later.

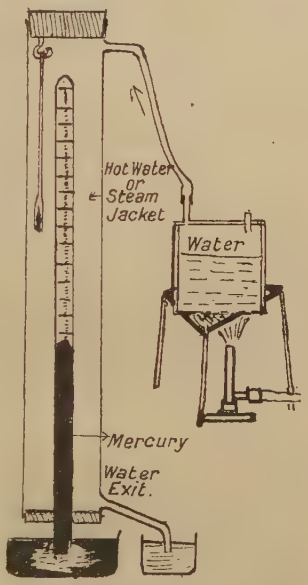

Fig. 28.—Aqueous vapour tension apparatus.

Dalton's Law of Partial Pressures (see also Henry's Law).—When a mixture of gases occupies a given space each gas is apportioned equally throughout the whole space. Furthermore, the pressure exerted by the mixture is the same as if each constituent alone occupied the space. The law of Dalton states that *the pressure exerted by the mixture is the sum of the pressures due to the individual constituents comprising the mixture.* Thus if we consider air, which is a mixture of, *roughly*, twenty volumes of O_2 and eighty volumes of N_2, the pressure exerted by the oxygen is only one-fourth of that exerted by the nitrogen, so that the atmospheric pressure is practically the sum of the individual pressures set up by these separate constituents. Again, 1 c.c. of water at ordinary temperature and pressure will dissolve 0·041 c.c. of O_2 and only 0·02 c.c. of N_2, so that if the air in the approximate proportions given above were dissolved in water, one litre,

(1000 c.c.), would contain $\frac{20}{100} \times \frac{1000}{1} \times \cdot 041 = 8{\cdot}2$ c.c. of O_2,

and $\frac{80}{100} \times \frac{1000}{1} \times \cdot 02 = 16{\cdot}0$ c.c. of N_2.

Then in the evaporated or boiled-out gas the partial pressure due to oxygen, which we see is twice as soluble as nitrogen, will be that due to $\frac{2}{2+4\cdot1}\times\frac{100}{1}=32\cdot8$ per cent. of oxygen. Hence the law is frequently modified thus: "*the solubility of a gas contained in a mixture of gases is proportional to its partial pressure.*" From this it will be seen that the weight of a cubic foot of moist air is the sum of the partial pressures of the dry air portion and the aqueous vapour portion of the cubic foot. The barometrical pressure for a complete cubic foot of dry air is greater than that for moist air, since aqueous vapour is not so dense as air. In other words, moist air is lighter than dry air.

We will now consider the case of two shafts, downcast and upcast, ventilating a mine, the absolute temperature of the air in each being T_1 and T_2 respectively. Under neither of these temperatures can the air expand laterally owing to the sides of the shaft, therefore the effect is to elongate a unit or column of air in the vertical line of the shafts. This expansion for a cubic foot of downcast air will be in the ratio $\frac{T_2}{T_1}$, which will be the height of the cubic foot when in the upcast shaft, and if the depth of the shafts be D feet then the total will be $D\frac{T_2}{T_1}$ feet, *i.e.* the column of air at upcast temperature will have to be $D\frac{T_2}{T_1}$ feet long in order to balance a column D feet long in the downcast. Since both the shafts are D feet deep the difference of the column effect between them is $D\frac{T_2}{T_1}-D\frac{T_1}{T_1}$ or $D\left(\frac{T_2-T_1}{T_1}\right)$ feet. This is obviously a direct measure of the ventilating pressure, and is known as the *motive column*, thus—

$$M=D\frac{(T_2-T_1)}{T_1} \text{ in feet of upcast air} \quad . \quad . \quad (57)$$

Commencing the argument in terms of the upcast and pursuing it from a standpoint of shrinkage when in the downcast, we get,

$$M=D\frac{(T_2-T_1)}{T_2} \text{ in feet of downcast air} \quad . \quad . \quad (58)$$

The pressure per square foot producing ventilation can be got by multiplying the weight of a cubic foot of air in the downcast

by the motive column in terms of downcast air, or by the product of the weight of a cubic foot in the upcast and the motive column in the upcast. The ventilating pressure so obtained will be the same in each case.

In the above formulæ, (57) and (58), the absolute temperature numerators give the same difference as between ordinary temperatures read from the thermometer. Taking T for upcast and t for downcast (ordinary temperatures), we can now modify the above formulæ to the following well-known forms :—

$$M = D\frac{(T - t)}{459 + t} \text{ in feet of upcast air} \quad . \quad . \quad (59)$$

$$M = D\frac{(T - t)}{459 + T} \quad ,, \quad ,, \quad \text{downcast air} \quad . \quad . \quad (60)$$

which result will be of use to us later.

Henry's Law.—*The weight of gas dissolved by water is directly proportional to the pressure* (this does not apply to very soluble gases). Oxygen, under a pressure of 5 atmospheres, *i.e.* $14{\cdot}7 \times 5 = 73{\cdot}5$ lbs. per sq. in., is five times as soluble as when under the ordinary pressure of 1 atmosphere. Certain mining fire-extinguishers employ this principle. They are made by charging a body of water with carbon dioxide and other non-inflammable gases under a fairly high pressure, which, when released, causes a strong jet and effervescence or fizzing of the liquid. The example of soda-water will be familiar to all. The mass of gas absorbed being in direct proportion to the pressure, and also by Boyle's Law the density of the gas is in the same ratio, the volume of a gas absorbed in a liquid will be constant. Hence the statement which is frequently used, " the volume of a gas dissolved in a liquid is independent of the pressure " (except for very soluble gases like ammonia, hydrochloric acid, etc.). Thus in any space occupied by two or more gases the relation between any partial pressure p_1, due to a single constituent of volume, v_1, and the pressure of the whole mixture, p_2, of volume, v_2, is equal to the relation between their respective weights, d_1, d_2, and gas constants R_1, R_2 (Table IV.), thus (by Dalton's Law also)—

$$\frac{p_1}{p_2} = \frac{v_1}{v_2} = \frac{d_1 R_1}{d_2 R_2}.$$

If one of the constituents is a vapour of molecular weight, m_1,

then $\frac{\text{weight of vapour}}{\text{weight of gas}} = \frac{m_1}{m_2}$ (m_2 is the molecular weight of the gas).

Graham's Law of Diffusion.—If in a vessel two or more gases exist, which are not uniformly distributed throughout the available space, they will tend to apportion themselves out regularly as stated when dealing with the law of partial pressures. The molecules of each gas will move from points of higher to points of lower density or pressure until the gaseous equilibrium or diffusion is complete. The process is rapid and unaffected by gravity. The fact that any odorous substance like coal-gas can be smelt at a considerable distance away from the source at which it is liberated, even in still air, can only be attributed to diffusion, and Graham, who first established the law relating to this property of gases, found that *the rates of diffusion of gases are inversely proportional to the square roots of their densities.*

E.g. the density of hydrogen, H_2, is 1; of methane, CH_2, 8; of carbon dioxide, CO_2, 22; of CO, 14; and of sulphuretted hydrogen, H_2S, 17.

∴ The rates of diffusion of these gases are in the ratio

$$\frac{1}{\sqrt{1}} : \frac{1}{\sqrt{8}} : \frac{1}{\sqrt{22}} : \frac{1}{\sqrt{14}} : \frac{1}{\sqrt{17}},$$

or

$$1 : \frac{1}{2{\cdot}83} : \frac{1}{4{\cdot}7} : \frac{1}{3{\cdot}7} : \frac{1}{4{\cdot}1} \text{ approx.}$$

That is to say that hydrogen diffuses

2·83	times	as	quickly	as methane.
4·7	,,	,,	,,	carbon dioxide.
3·7	,,	,,	,,	carbon monoxide.
4·1	,,	,,	,,	sulphuretted hydrogen.

Thus the presence of firedamp, which is largely composed of CH_4, will be more quickly discerned than the presence of most of the other mine gases.

It is very important to note here that the segregation of a gas from a uniform mixture of gas and air, so that there is a richer mixture near the roof than lower down, is an impossibility, because the law of diffusion of gases does not admit of it. Where a richer stratum of gas is found nearer the roof than the floor it is due to gas being liberated at a greater rate than it is being

diffused, and similarly for black-damp near the floor the same reasoning holds.*

Bernouilli's Theorem.—A fluid in motion possesses energy in three forms, viz.—

i. Potential energy, h, due to its height above some datum.

ii. Kinetic energy, $\frac{v^2}{2g}$, due to its motion.

iii. Pressure energy, $\frac{p}{d}$, due to the pressure head.

E.g. suppose a liquid of density d lbs. per cubic foot is contained in a reservoir, A (fig. 29), whose free horizontal surface is kept at

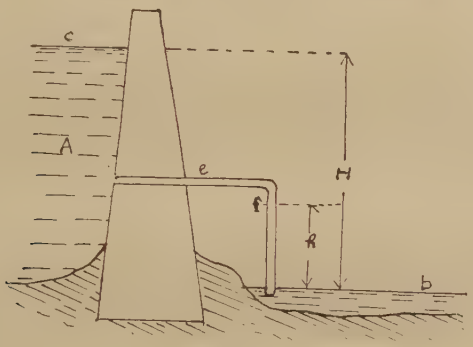

Fig. 29.

a constant height, H ft., above any datum (in most mining and engineering problems atmospheric pressure is taken as the datum pressure, so that a fluid, say water, discharging under atmospheric pressure in a parallel jet is taken as having no energy due to this source). Let the fluid be discharged by a connecting pipe, e, to the reservoir, b. Now imagine a point, c, in the surface and a point, f, in the connecting pipe h ft. above the datum. Let v ft. per second be the velocity of a particle at f while the pressure is p lbs. per square foot, and let a be the cross-sectional area of the pipe in square feet. Now if a piston be imagined to move with the water at f, a fall of 1 ft. means

* See also Prof. Wheeler's papers, "Ignition of Firedamp" and "Firedamp Explosions" (discussion thereon), and Prof. Granville Poole, *The Colliery Year Book*, 1923.

that $1 \times p \times a$ or pa ft.-lbs. of work are being done on the piston while "a" cubic feet of water leaves the pipe, or we may say

pa, ft.-lbs. of work for a cubic feet,
i.e. p, ft.-lbs. of work for each cubic foot.

In other words, since 1 cub. ft. contains d lbs., $\frac{p}{d}$ ft.-lbs. of work are done for each pound of liquid leaving the pipe, and taking atmospheric pressure as zero, the energy of the water at c is entirely potential, due to the head H ft. Then it follows that the energy of the water at f must exist as (1) the potential energy h, due its height, above a given datum; (2) kinetic energy, $\frac{v^2}{2g}$, also a head where v is the velocity acquired by a body falling freely from the height given; and (3) the pressure energy, $\frac{p}{d}$, which is the height of a column of water to produce a pressure p lbs. per square foot at its base, and is therefore called the pressure head. Thus if fluid flows from one point to another the cumulative effect due to these three heads, or the total energy due to position, motion, and pressure, neglecting friction losses, is

$$H = h + \frac{v^2}{2g} + \frac{p}{d} \quad . \quad . \quad . \quad . \quad (61)$$

This may be expressed for a flow between two points 1 and 2 as

$$h_1 + \frac{v^2_1}{2g} + \frac{p_1}{d} = h_2 + \frac{v^2_2}{2g} + \frac{p_2}{d} \quad . \quad . \quad . \quad (62)$$

where the subscripts indicate the points of flow to which the terms apply. If loss of head h_0 exists, due to friction against the surfaces, bends, and eddying caused by the effect of varying areas on the current velocity, etc., then (62) becomes

$$h_1 + \frac{v^2_1}{2g} + \frac{p_1}{d} = h_2 + \frac{v^2_2}{2g} + \frac{p_2}{d} + h_0 \quad . \quad . \quad . \quad (63)$$

E.g. air weighing 0·076 lb. per cubic foot moves with a velocity of 60 ft. per second at the tips of a fan runner. At a point of the same height in the fan chimney the velocity of the air is reduced to 20 ft. per second. What is the change in pressure per square foot assuming no friction ?

by (62) $$h_1 + \frac{v^2_1}{2g} + \frac{p_1}{d} = h_2 + \frac{v^2_2}{2g} + \frac{p_2}{d}$$

but $h_1 = h_2$ (given)

then $$\frac{p_2 - p_1}{d} = \frac{v^2_1 - v^2_2}{2g}$$

$$\therefore \quad p_2 - p_1 = d\,\frac{(v^2_1 - v^2_2)}{2g}$$

$$= \frac{0{\cdot}076\ (3600 - 400)}{64}.$$

Change in pressure = 3·8 lbs. per square foot.

or $$\frac{p}{d} = \frac{3{\cdot}8}{0{\cdot}076} = 50 \text{ ft. of air column change of head.}$$

The Law of Venturi.—*If the cross-sectional area of a pipe diminishes, the pressure of a fluid flowing in it decreases as the velocity increases, and vice versa.*

The Venturi Meter. — By considering together these three

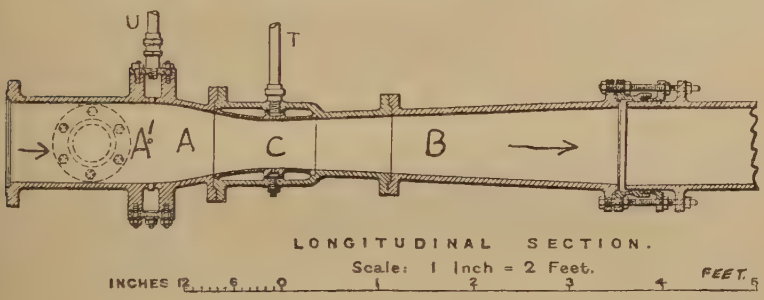

Fig. 30.—The Venturi meter.

heads, potential, kinetic, and pressure, or the total energy, and neglecting friction, we may state that the total energy at *any* cross-section is the sum of these three heads. The sum of the velocity and pressure heads always remain constant (disregarding friction), and therefore the higher the velocity due to a restricted cross-section the lower the pressure head. This principle is adopted in the Venturi Tube * (fig. 30). It consists of two cone pieces, A and B, with their apices truncated and joined by a throat section, C. When a fluid passes in the direction of the arrow its velocity increases at C, and, on flowing through, it passes into the diverging area for B. This arrangement guides the stream lines evenly to the throat, and, finally, on to the low-

* J. L. Hodgson, *Trans. Inst. M.E.*, 1922.

pressure area, B, which usually has the same cross-section, and, therefore, shows the same pressure as A′. If a manometer or water-gauge, U, be attached to the full area, A′, and another, T, to the throat, C, the difference of pressure registered by them will be in accordance with Bernouilli's theorem. The velocity being increased at the throat the pressure will fall, which fall will be proportional to the square of the velocity, and, consequently, *the velocity will be proportional to the square root of the pressure or head.* Since this result is incorporated in the most important representation of the laws of mine ventilation, *i.e.* " Atkinson's Formula," it behoves us at this stage to pursue it somewhat

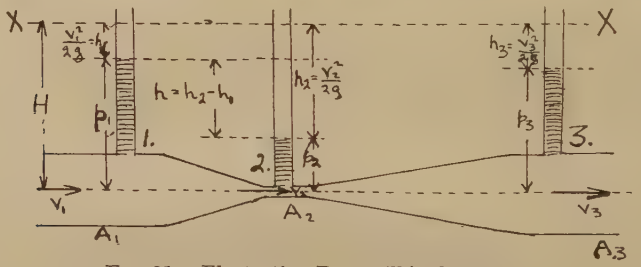

FIG. 31.—Illustrating Bernouilli's theorem.

further. Referring to fig. 31, where X, X is a datum line, or free level of a fluid, as in a reservoir, we may consider 1, 2, and 3 to be manometers. The full bore where manometer 1 is placed is of cross-section A_1 sq. ft., the throat bore for 2, A_2 sq. ft., and that at final bore for 3 is A_3 sq. ft., equal to A_1. As the weight of fluid which must pass each section is the same the total energy is by (62)

at gauge 1, $$W\left(h_1+\frac{v_1^2}{2g}+\frac{p_1}{d}\right) \quad . \quad . \quad . \quad (64)$$

at gauge 2, $$W\left(h_2+\frac{v_2^2}{2g}+\frac{p_2}{d}\right) \quad . \quad . \quad . \quad (65)$$

at gauge 3, $$W\left(h_3+\frac{v_3^2}{2g}+\frac{p_3}{d}\right) \quad . \quad . \quad . \quad (66)$$

and so on for any number of gauges and throats, provided we assume a perfect fluid, *i.e.* frictionless.

Now at pipe section 1, by (10), P.E. lost = K.E. gained, or

$$Wh_1=\frac{Wv_1^2}{2g} \therefore h_1=\frac{v_1^2}{2g} \quad . \quad . \quad . \quad (67)$$

Again, by (10) at section 2 (throat)—

$$W h_2 = \frac{W v_2^2}{2g} \text{ or } h_2 = \frac{v_2^2}{2g} \quad \text{(see fig. 31).} \quad (68)$$

Note that the heads are measured down from the datum line and the pressures up from the line of mid-flow. Now, still neglecting loss of head h_0, due to friction, etc., we see that the initial loss of head h_1 is that which creates the initial flow v_1. The loss of head h between the two points 1 and 2 is equal to the pressure fall between these points. This is called the "Venturi Head," and it clearly illustrates that pressure and head are synonymous terms. The fall in pressure from p_1 to p_2 is due to an increase in velocity from v_1 on entry to v_2 at the throat. This LOSS OF HEAD, h, will be seen to vary as the velocities v_1 and v_2, which in turn vary inversely with respect to the areas A_1 and A_2, thus verifying the theorem of Bernouilli. Hence—

$$\text{Change of head, } h = (h_2 - h_1) = \left(\frac{v_2^2}{2g} - \frac{v_1^2}{2g}\right) = \frac{v_2^2 - v_1^2}{2g} = p_1 - p_2,$$

and since $$h_2 = h + h_1 = \frac{v_2^2}{2g},$$

$$\therefore \quad v_2 = \sqrt{2g\left(h + \frac{v_1^2}{2g}\right)} \quad . \quad . \quad . \quad . \quad (69)$$

Knowing also that these velocities vary inversely as the areas A_1 and A_2, *i.e.* $v_1 = \left(\frac{A_2}{A_1}\right) v_2$ and $v_2 = \left(\frac{A_1}{A_2}\right) v_1$, it can be shown that the velocity at the throat is equal to

$$v_2 = \left(\frac{A_1}{\sqrt{A_1^2 - A_2^2}}\right)\sqrt{2gh} \quad . \quad . \quad . \quad (70)$$

which result is very useful when dealing with ventilation regulators. As h is often expressed in inches of water gauge we can state that the difference in water gauge due to velocity is proportional to the square of the velocity, and therefore this velocity varies as the square root of the above water gauge, and inversely as the cross-sectional area. These facts will form the foundation of our discussion of the dynamical and static water gauges later.

We have said nothing about friction losses in the Venturi

meter, since we are chiefly concerned with its principle. Mr J. L. Hodgson * gives it as $\frac{1}{8}$th.

International Atomic Weights.

(O = 16.)

Table V.—(*The Elements in the Order of the Atomic Weights.*)

Name.	Symbol.	Atomic weight.	Name.	Symbol.	Atomic weight.
Hydrogen	H	1·008	Ruthenium	Ru	101·7
Helium	He	4·00	Rhodium	Rh	102·9
Lithium	Li	6·94	Palladium	Pd	106·7
Glucium (Beryllium)	Gl or Be	9·1	Silver	Ag	107·88
Boron	B	10·9	Cadmium	Cd	112·4
Carbon	C	12·005	Indium	In	114·8
Nitrogen	N	14·01	Tin	Sn	118·7
Oxygen	O	16·00	Antimony	Sb	120.2
Fluorine	F	19·0	Iodine	I	126·92
Neon	Ne	20·2	Tellurium	Te	127·5
Sodium	Na	23·00	Xenon	Xe	130·2
Magnesium	Mg	24·32	Cæsium	Cs	132·81
Aluminium	Al	27·1	Barium	Ba	137·37
Silicon	Si	28·3	Lanthanum	La	139·0
Phosphorus	P	31·04	Cerium	Ce	140·25
Sulphur	S	32·06	Praseodymium	Pr	140·9
Chlorine	Cl	35·46	Neodymium	Nd	144·3
Potassium	K	39·10	Samarium	Sa	150·4
Argon	A	39·88	Europium	Eu	152·0
Calcium	Ca	40·07	Gadolinium	Gd	157·3
Scandium	Sc	44·1	Terbium	Tb	159·2
Titanium	Ti	48·1	Dysprosium	Dy	162·5
Vanadium	V	51·0	Holmium	Ho	163·5
Chromium	Cr	52·0	Erbium	Er	167·7
Manganese	Mn	54·93	Thulium	Tm	168·5
Iron	Fe	55·84	Ytterbium	Yb	173·5
Nickel	Ni	58·68	Lutecium	Lu	175·0
Cobalt	Co	58·97	Tantalum	Ta	181·5
Copper	Cu	63·57	Tungsten	W	184·0
Zinc	Zn	65·37	Osmium	Os	190·9
Gallium	Ga	69·9	Iridium	Ir	193·1
Germanium	Ge	72·5	Platinum	Pt	195·2
Arsenic	As	74·96	Gold	Au	197·2
Selenium	Se	79·2	Mercury	Hg	200·6
Bromine	Br	79·92	Thallium	Tl	204·0
Krypton	Kr	82·92	Lead	Pb	207·2
Rubidium	Rb	85·45	Bismuth	Bi	208·0
Strontium	Sr	87·63	Niton—(Radium emanation)	Nt	222·0
Yttrium	Y	88·7	Radium	Ra	226·0
Zirconium	Zr	90·6	Thorium	Th	232·4
Niobium (Columbium)	Nb or Cb	93·1	Uranium	U	238·2
Molybdenum	Mo	96·0			

* "The Metering of Large Volumes of Air," *Trans. Inst. M.E.*, vol. lxii, 1922.

The Law of Definite or Constant Proportions.—As a result of a vast number of researches by competent authorities it has been established that a *given chemical compound, in whatever manner it is prepared, always contains its elements in the same proportion by weight.* We should find by taking a quantity of water prepared in the laboratory by combining one part by weight of hydrogen with eight parts by weight of oxygen that its composition would be identical with that of pure water from any part of the world and from any source. This law applies to all compounds.

Chemical Equivalents.—In order to understand the nature of chemical combination fully we must have a standard whereby the amounts by weight of substances combining may be compared. This is done by taking hydrogen, the lightest substance known, as the standard basis for comparison. *The weight of an element which will combine with or replace one part by weight of hydrogen is called its equivalent weight.* Now every element cannot combine with hydrogen, so the equivalent weight of many elements must be found by their method of combining with an element which *does* combine with hydrogen. Oxygen enters into combination with most elements, so if we know the equivalent weight of oxygen we can find by proportion that of any element which will combine with or displace a certain amount of oxygen. On consulting the atomic weights the following facts will impress the above statements upon the mind :—

Water, H_2O, is composed of 1 part of hydrogen and 8 parts of oxygen by weight.

Methane, CH_4, is composed of 1 part of hydrogen and 3 parts of carbon by weight.

Sulphuretted hydrogen, H_2S, is composed of 1 part of hydrogen and 16 parts of sulphur by weight.

Carbon dioxide, CO_2, is composed of 3 parts of carbon and 8 parts of oxygen by weight.

From this we see that the equivalents of hydrogen, carbon, sulphur, and oxygen are 1, 3, 16, and 8 respectively, so that we can establish the equivalent of any element by finding its equivalent towards oxygen, or, in fact, towards any body whose equivalent to hydrogen is known.

The Law of Multiple Proportions.—There are many cases in which an element will combine with another element in more than one proportion by weight, and a few examples are as follows :—

Case 1.

In carbon monoxide, CO, 16 parts of oxygen combine with 12 of carbon by weight.

In carbon dioxide, CO_2, 32 parts of oxygen combine with 12 of carbon by weight.

Case 2.

In nitrous oxide, N_2O, 16 parts of oxygen combine with 28 of nitrogen by weight.

In nitric oxide, NO, 32 parts of oxygen combine with 28 of nitrogen by weight.

In nitrogen trioxide, N_2O_3, 48 parts of oxygen combine with 28 of nitrogen by weight.

In nitrogen peroxide, NO_2, 64 parts of oxygen combine with 28 of nitrogen by weight.

In nitrogen pentoxide, N_2O_5, 80 parts of oxygen combine with 28 of nitrogen by weight.

In the first case the ratio of the combining weights of oxygen with carbon is 1 : 2.

In the second case the ratio of the combining weights of oxygen with nitrogen is 1 : 2 : 3 : 4 : 5.

There are many other examples.

We see from the above examples that *if two elements*, A *and* B, *combine together in more than one proportion the weights of* A *which combine with a fixed weight of* B *bear a simple relation to each other.*

Dalton's Atomic Theory.—Dalton explained the above facts by assuming that all matter is made up of extremely minute indivisible particles or atoms, and that for the same element the atoms are all identical in weight, different elements having different atoms. He also supposed that chemical combination is due to the grouping together of the atoms to form compound atoms. By this theory it will be seen that the size of a body in no way affects the structure, and we are thus able by chemical analysis to find not only the percentage composition of bodies but also the relative combining weights. This has been done for all the known elements, taking hydrogen, the lightest substance known, as the unit of weight for comparison, and Dalton's atomic theory has been found to harmonise with all the facts.

Gay Lussac's Law of Volumes.—This important law states that "*volumes of gases which combine together chemically bear a simple relation to one another and the resulting product, if a gas*, e.g.—

(*a*) 2 volumes of hydrogen plus 1 of oxygen give 2 volumes of water (steam).

(*b*) 1 volume of hydrogen plus 1 of chlorine give 2 volumes of hydrochloric acid.

In (*a*) the ratio is 2 : 1 : 2.

In (*b*) the ratio is 1 : 1 : 2. These, or equally simple relations, hold for all combinations of gases.

Avogadro's Hypothesis.—Under equal conditions of temperature and pressure all gases, however dissimilar their properties, behave in the same way as was seen in discussing Boyle's and Charles' Laws. Avogadro explained this by suggesting that equal volumes of all gases, under the same conditions of temperature and pressure, contain an equal number of particles. When one

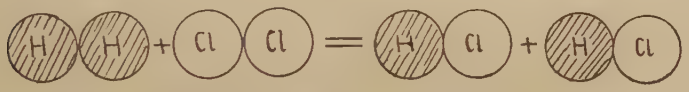

Fig. 32.

volume of hydrogen combines with one of chlorine we get two volumes of hydrochloric acid gas. We may therefore say that if n particles of chlorine combine with n particles of hydrogen, we get $2n$ particles of hydrochloric acid gas. It is evident in the former case that the particle of hydrogen cannot be 1 atom, since each of the two particles of hydrochloric acid gas must contain at least one atom of hydrogen. Avogadro explained this by making a distinction between the ultimate atom of an element and the physical unit or molecule which may be made up of many atoms.

The Atom is the smallest particle of an element which can exist and take part in chemical change.

The Molecule, which consists of two or more atoms, is the smallest particle of an element or compound capable of existing alone and still possessing the properties of the element or compound.

Avogadro therefore, by assuming the existence of such particles, destroyed the above discrepancy, and his hypothesis that *equal volumes of all gases, both elementary and compound, under the same conditions of temperature and pressure, contain the same numbers of molecules*, is verified by all the facts. This is expressed diagrammatically in fig. 32.

Since hydrogen is taken as the unit of weight, the *atomic weight* of an element is the relation which the weight of an atom of that

element bears to the weight of an atom of hydrogen. The *Molecular Weight* of any element or compound is the relation which the weight of a molecule of an element or compound bears to the weight of an atom of hydrogen, and as it may be shown that there are two atoms in the hydrogen molecule, its atomic weight being one, its molecular weight is, then, two. The molecule of most gaseous elements contains two atoms.

In the list of atomic weights supplied on p. 56 it will be noted that the atomic weight of hydrogen is taken as 1·008; this enables oxygen to be placed as 16, which brings out a bigger number of the atomic weights of the other elements into round figures, so facilitating computation. For our purpose we will here use only approximate figures.

Density of Gas.—From the above hypothesis of Avogadro we may now state the following:—

$$\text{The relative density of a gas} = \frac{\text{Weight of 1 volume of the gas}}{\text{Weight of 1 volume of hydrogen}}$$

$$i.e. \quad = \frac{\text{Weight of } n \text{ molecules of the gas}}{\text{Weight of } n \text{ molecules of hydrogen}}$$

or

$$= \frac{\text{Weight of 1 molecule of the gas}}{\text{Weight of 1 molecule of hydrogen}} = \frac{\text{Molecular weight of the gas}}{2}$$

$\therefore$ The molecular weight of a gas $= 2 \times$ density of the gas, or *the density of a gas is half its molecular weight.*

It has been well established that 1 litre (1000 cub. cms.) of hydrogen weighs 0·0896 grams at N.T.P. (normal temperature and pressure, *i.e.* 0° C. and 760 mm.), so that 2 grams must occupy 22·24 litres. Now seeing that 22·24 litres of hydrogen weigh 2 grams, and that the relative density of oxygen is 16, then 22·24 litres of oxygen must weigh $2 \times 16 = 32$ grams; and since the atomic weight of oxygen is 16, and the molecule of oxygen contains 2 atoms, the molecular weight of oxygen is therefore 32. It is seen that the molecular weight of oxygen in grams occupies 22·24 litres at N.T.P.

Again, since the relative density of chlorine is 35·5, 22·24 litres of chlorine at N.T.P. weighs $2 \times 35{\cdot}5 = 71$ grams. Furthermore, the atomic weight of chlorine is 35·5, and 1 molecule of chlorine contains 2 atoms, therefore the molecular weight of chlorine is 71. It is thus seen that the molecular weight in grams of chlorine also occupies 22·24 litres at N.T.P. This

can be generally stated thus: *the molecular weight of any gas, in grams, occupies a space of 22·24 litres at N.T.P.*

$$1 \text{ litre} = \frac{1}{28{\cdot}3} \text{ cub. ft. or } 0{\cdot}0353 \text{ cub. ft.} = 61 \text{ cub. in.}$$

$$1 \text{ gram} = \frac{1}{453{\cdot}6} \text{ lbs. or } 0{\cdot}0022 \text{ lbs.} \qquad = 15{\cdot}43 \text{ grains.}$$

The Specific Gravity of a Gas is the ratio of the weight of a given volume of the gas to that of an equal volume of air under the same conditions of temperature and pressure.

Petit and Dulong's Law.—This law states that *the atomic weight of an element multiplied by its specific heat in the solid state is approximately* 6·4.

E.g. The atomic weight of lead is 207·10.
The specific heat of lead is 0·031.

$$207{\cdot}10 \times 0{\cdot}031 = 6{\cdot}42.$$

This often forms a check on the atomic weight of an element, since it will be seen that the atomic weight is inversely proportional to its specific heat, or

$$\text{atomic weight} = \frac{1}{\text{sp. ht.}} \times 6{\cdot}4 \qquad . \qquad . \quad (71)$$

Acids and Alkalies.—An acid may exist in any of the three states of matter. Acids are sour in taste, always contain hydrogen, turn blue litmus red, and the hydrogen is replaceable by a metal. The oxides produced by burning metals in oxygen may either turn blue litmus red, when they are called acid oxides, or may have no effect upon it, when they are said to be basic oxides. Among the common and important acids the chief are sulphuric acid, H_2SO_4, nitric acid, HNO_3, and hydrochloric acid, HCl.

Alkalies turn red litmus blue and have the reverse effect of acids. It will be seen that acids and alkalies neutralise each other. The result of this neutralisation is a salt which has neither acid nor alkali properties.

A **normal solution** of any substance is made by dissolving the gram equivalent weight of the substance (*i.e.* the amount in grams which corresponds to one gram of hydrogen) in sufficient water to make one litre of solution. The equivalent of H_2SO_4 is 49, thus

$$Zn + H_2SO_4 = ZnSO_4 + H_2$$
$$\overbrace{2 + 32 + 4 \times 16} \quad : \quad \overbrace{2}$$

i.e. 98 : 2 or 49 : 1, which means that there are 49 grams of sulphuric acid per litre in a normal solution of it. The strength of a normal solution is often indicated by the letter N, and that of a solution one-tenth as strong or a **decinormal solution** by $\frac{N}{10}$; thus $\frac{N}{10}.H_2SO_4$ is a decinormal solution of sulphuric acid, and contains 4·9 grams of the acid per litre or 0·0049 grams per c.c. Any solution the strength of which is known is called a **standard solution.**

When solutions of acids and alkalies are gradually added to one another there will come an instant when they will neutralise each other by just the addition of another drop. To find out exactly where this takes place substances called indicators are put into the solution, the characteristic coloration noted, and the amount needed to produce it. These indicators show, then, by their colour effect the amount of acid or alkali in excess, and therefore are a measure of the strength of the solution. This process is known as *titration*.

We are now in a position to distinguish between the properties of compounds and mixtures.

Compounds.	*Mixtures.*
1. The component elements combine in definite proportions. *e.g.* The composition of nitrous oxide is always 28 of nitrogen and 16 of oxygen.	1. The component elements may exist in any proportion. *e.g.* The proportion of nitrogen and oxygen in the atmosphere, which is a mixture, varies with the altitude and other causes.
2. The components of which a compound consists do not exhibit any of the physical properties they possessed as elements. *e.g.* When iron and sulphur combine chemically they produce iron sulphide which will not dissolve in carbon disulphide, as does sulphur, nor can the iron be detected by the microscope or any other physical means.	2. The components retain all their physical properties and can be separated from the mixture mechanically. *e.g.* The sulphur in the case of a mixture of iron and sulphur can be dissolved in iron disulphide and the iron removed by a magnet.
3. When elements combine, there is always either an evolution or absorption of heat.	3. When the components are mixed no heat is given out or taken up.

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CHAPTER III.

THE ATMOSPHERE AND MINE GASES.

The air, though a mechanical mixture, is remarkably constant in composition, varying only very slightly from place to place and time to time under natural conditions. It is composed chiefly of

Oxygen	20·93	per cent.	by	volume.
Nitrogen	78·10	,,	,,	,,
Argon	0·93	,,	,,	,,
Carbon dioxide	0·03	,,	,,	,,

In addition we get small traces of ammonia, due to the putrefaction of organic matter containing nitrogen, but as this gas is very soluble in water it is easily cleared, especially by rain. Small but variable quantities of aqueous vapour exist, and also helium, neon, krypton, and xenon, but these four being found in such very minute quantities, and also being, like argon, devoid of chemical properties, need no further mention for our purpose. Indeed we may for all intents and purposes look upon air as consisting of

21 parts of oxygen by *volume*.
79 parts of nitrogen by *volume*.

and, knowing their relative densities, we see that they exist also in the ratio $\frac{79}{21} \times \frac{14}{16}$ or 1 : 3·3, *i.e.* $\frac{100}{1+3\cdot3} = 23$ approx. for oxygen.

i.e. 23 parts of oxygen by *weight*
77 parts of nitrogen by *weight* } approx.

Oxygen, O_2, is an element of great importance, being about one-fifth of the atmosphere by volume. It is essentially necessary for combustion and respiration. It has no colour, taste, or smell, is very active chemically, and forms a great number of

compounds. It may be prepared in many ways by heating numerous compounds or by the fractional distillation of liquid air. The simplest derivatives of oxygen are called oxides, the non-metallic oxides having acid properties with water, and the metallic ones basic properties (opposed to acids). Its great chemical activity and number of compounds has accounted for its being adopted as the standard 16 for atomic weights, which makes hydrogen 1·008 in the new notation, and not 1 as suggested first by Dalton—see the list of atomic weights, Table V. Its specific gravity is 1·11 (air = 1). Oxygen is allotropic, *i.e.* capable of existing in another form, and in the other form is known as ozone, O_3, which has powerful oxidising properties. Ozone is triatomic, *i.e.* it contains three atoms in the molecule, whilst oxygen is diatomic. Water is $\frac{1}{9}$rd oxygen by volume and $\frac{8}{9}$ths by weight.

The influence of oxygen on combustion and life will be appreciated from the following table :—

With air containing 16·9 to 17 per cent. of oxygen a soft candle flame is extinguished.

With air containing 16·9 per cent. of oxygen a petroleum flame is extinguished.

With air containing 11·5 per cent. of oxygen a coal-gas flame is extinguished.

With air containing 17.5 per cent. of oxygen a methane flame is extinguished.

With air containing 12·5 per cent. of oxygen breathing is difficult, even at sea-level.

With air containing 10 per cent. of oxygen there is great danger to life.

With air containing $7\frac{1}{2}$ per cent. of oxygen death occurs.

Oxygen varies from 20·4 to 21 per cent. of the atmosphere by volume.

Nitrogen, N_2, is chiefly characterised by its inertness or inactivity. It is colourless, tasteless, odourless, and incapable of supporting combustion or life. It combines with oxygen under the influence of the electric arc, forming some of the oxides of nitrogen mentioned in the previous chapter: it also forms, at high temperature, certain metallic nitrides. It is generally prepared from its compounds, but it can be fixed from the atmosphere by using caustic potash to absorb the carbon dioxide, calcium chloride the aqueous vapour, and copper or phosphorus the oxygen. The residue is often 99 per cent. nitrogen. Like oxygen

it can be liquefied and solidified by the application of extreme cold and high pressure. The function of nitrogen in the atmosphere is largely that of a diluent of the oxygen. Life in an atmosphere of oxygen would be very short and gay. Nitrogen plays an important rôle in the life of plants which derive their nitrogen from the soil. It has a diatomic molecule, and its atomic weight is 14·01, in terms of oxygen at 16·00, and therefore it has a specific gravity of 0·973 (air = 1).

Carbon Dioxide, CO_2, also called carbonic acid or carbonic anhydride, although present in the atmosphere only in minute proportions (about 3 or 4 volumes in 10,000 volumes of air), is of great importance. It is prepared by the combustion of carbon in oxygen or from the metallic carbonates by means of dilute acids. It is fairly soluble (water absorbing more than its own volume of it at ordinary temperature), but is incapable of supporting combustion or life. Plants assimilate it, at the same time liberating a nearly equal quantity of oxygen. Its presence in the atmosphere is due chiefly to the respiration of man and the animals, as well as the combustion of organic matter. It is also discharged into the atmosphere near stagnant pools and from fissures. It accumulates in crowded rooms owing to respiration, and also in localities where much coal is burnt, *i.e.* near big towns. Its specific gravity is 1·53 (air = 1).

Water Vapour. This constituent of the air varies considerably with the temperature since the atmosphere can only take up a certain quantity of it at a given temperature, the air being then saturated. The vapour pressure of the water at that temperature is the measure of the quantity of water capable of being absorbed. This vapour pressure of water rises rapidly with the temperature, as stated in the previous chapter, and consequently the quantity of water present in the air depends on the temperature, hence the reason why winter air is the driest air.

The amount of aqueous vapour present in the air depends on

(*a*) the temperature; this is the most important factor.
(*b*) the proximity of large bodies of water.
(*c*) the configuration of the land.
(*d*) the atmospheric pressure to a small extent.

Aqueous vapour is less dense than air, therefore moist air is lighter than dry air, which fact is influenced greatly by the temperature, and in some small measure by the pressure. See Table VI.

TABLE VI.—*Influence of Aqueous Vapour on the Weight of Air.*

Temp. in degs. F.	Water Vapour.		Weight in lbs. per cub. ft. of dry air at various absolute pressures.			Elastic force in lbs. per sq. in. of the oxygen-nitrogen portion, only, of saturated air at various absolute pressures.			Weight in lbs. of the oxygen-nitrogen portion, only, in each cub. ft. of saturated air at various absolute pressures.			Weight in lbs. per cub. ft. of saturated air at various absolute pressures.		
	Elastic force in lbs. abs. per sq. in.	Weight in lbs. per cub. ft.	14·7 lbs. abs.	14 lbs. abs.	13 lbs. abs.	14·7 lbs. abs.	14 lbs. abs.	13 lbs. abs.	14·7 lbs. abs.	14 lbs. abs.	13 lbs. abs.	14·7 lbs. abs.	14 lbs. abs.	13 lbs. abs.
1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	13.	14.	15.
32	·089	·00030	·08073	·07688	·07140	14·611	13·911	12·911	·08024	·07610	·07091	·08054	·07670	·07121
35	·100	·00034	·08024	·07642	·07096	14·600	13·900	12·900	·07969	·07587	·07041	·08003	·07621	·07075
40	·122	·00041	·07944	·07566	·07025	14·578	13·878	12·878	·07878	·07500	·06959	·07919	·07541	·07000
45	·147	·00049	·07865	·07491	·06955	14·553	13·853	12·853	·07786	·07412	·06876	·07835	·07461	·06925
50	·178	·00059	·07788	·07417	·06888	14·522	13·822	12·822	·07694	·07323	·06793	·07753	·07382	·06852
55	·214	·00070	·07713	·07345	·06821	14·486	13·786	12·786	·07601	·07231	·06709	·07671	·07304	·06779
60	·256	·00083	·07639	·07275	·06755	14·444	13·744	12·744	·07505	·07141	·06621	·07588	·07224	·06704
65	·305	·00098	·07566	·07206	·06691	14·395	13·695	12·695	·07409	·07048	·06534	·07507	·07146	·06632
70	·363	·00115	·07495	·07138	·06628	14·337	13·637	12·637	·07310	·06953	·06443	·07425	·07068	·06558
75	·429	·00135	·07425	·07017	·06566	14·271	13·571	12·571	·07208	·06855	·06350	·07343	·06990	·06485
80	·505	·00157	·07356	·07006	·06506	14·195	13·495	12·495	·07104	·06753	·06253	·07261	·06910	·06410
85	·594	·00183	·07289	·06942	·06446	14·106	13·406	12·406	·06994	·06647	·06151	·07177	·06830	·06334
90	·696	·00213	·07223	·06879	·06388	14·004	13·304	12·304	·06881	·06537	·06046	·07094	·06750	·06259
95	·813	·00247	·07158	·06817	·06330	13·887	13·187	12·187	·06762	·06421	·05934	·07009	·06668	·06181
100	·946	·00285	·07094	·06756	·06273	13·754	13·054	12·054	·06638	·06300	·05817	·06923	·06583	·06102
105	1·098	·00328	·07031	·06696	·06218	13·602	12·902	11·902	·06506	·06172	·05693	·06834	·06500	·06021

(By the oxygen-nitrogen portion of the air is meant air free from other gases, water vapour, etc.)
Columns 2 and 3 are obtained from standard tables.*
Columns 4, 5, and 6 are obtained from formula (54).
,, 7, 8, and 9 ,, ,, by subtracting column 2 from the given absolute pressure of the saturated air.
,, 10, 11, and 12 ,, ,, ,, formula (52) multiplied by the percentage of the oxy-nitrogen portion in the air.
,, 13, 14, and 15 ,, ,, ,, adding column 3 to columns 10, 11, and 12.

* By permission of the South African Institution of Engineers.

The Air of Mines.—Mine atmospheres are subject to contamination with noxious gases given off by the strata and from other sources. The percentage of firedamp and other gases present depends very much upon the atmospheric pressure and temperature. The principal additional factors which affect the changes in mine air are—

i. the velocity of the air in the mine.
ii. the amount of coal with which the air comes into contact.
iii. the gaseous nature of the coal (coals vary in their yield of methane: many English coals yield over five hundred times their volume when heated to 100° C.).
iv. the nature of the coal respecting its power to react with oxygen.
v. the temperature and the wetness of the mine.

To ensure a fair margin of comfort and safety the C.M.A., 1911, stipulates the following provisions (sec. 29): "that for the purpose of this section a place shall not be deemed to be in a fit state for working or passing therein if the air contains (by volume) either

(*a*) Less than 19 per cent. of oxygen;
(*b*) More than $1\frac{1}{4}$ per cent. of carbon dioxide;

and an *intake* airway shall not be deemed to be kept normally free from *inflammable gas* if the average percentage of six samples taken by an inspector in the air current at intervals of not less than a fortnight exceeds one-quarter." The Act also states (sec. 67) that workmen must be withdrawn, and a place shall be deemed to be dangerous if the percentage of inflammable gas in the *general body* of the air in that place is found to contain $2\frac{1}{2}$ per cent. or upwards, or, if situated in a part of the mine worked by naked lights, $1\frac{1}{4}$ per cent. or upwards. There are also restrictions with regard to the use of safety lamps (sec. 32) and haulage on return airways in newly-opened-out mines (sec. 42 and Gen. Regs.), all of which indicate the possible fluctuation in the percentage of mine air constituents. Now apart from any amount needed to dilute deleterious gases, a grown person hard at work needs from 0·5 to 1·05 cub. ft. of air per minute, or one-third this amount when at rest.* This air if normal (*i.e.* of the composition given at the commencement of this chapter) will, after respiration,

* Trautwine.

contain $3\frac{1}{2}$ to 4 per cent. of carbon dioxide, 17 to $16\frac{1}{2}$ per cent. of oxygen, and about $79\frac{1}{2}$ per cent. of nitrogen.* This means roughly $\frac{1}{25}$th part of the air that has passed through the lungs consists of carbon dioxide, and, accepting the above quantity as needed by a man at work, each adult would exhale from 0·021 to 0·044 cub. ft. of CO_2 per minute. If this were the only source of carbon dioxide the ventilating current would have to provide from 10·5 to 22 cub. ft. of fresh air per minute, free of CO_2, for each person, but the atmosphere already contains approximately $3\frac{1}{2}$ parts of CO_2 in every 10,000, which will increase these figures to approximately 12 and 26 cub. ft. respectively per person per minute. Now on entering the mine there is the possibility of a further $1\frac{1}{4}$ per cent. of CO_2, the maximum permissible by sec. 29 C.M.A., 1911, so that the above method of dilution must be extended, and it is fairly usual to allow 200 to 250 cub. ft. of air per minute per person employed. Thus to dilute or render harmless any noxious gases produced, and also have a comfortable reserve in the mine, 200 to 500 times the volume of fresh air inhaled by each person must pass through the working places. In fiery mines this amount is often exceeded. A horse needs three times as much oxygen as a man, and an oil lamp 0·0188 cub. ft. of oxygen per minute.†

The Mine Gases. –The four chief gases met with in mines were formerly held to be carbon dioxide, CO_2; methane, CH_4; carbon monoxide, CO; and sulphuretted hydrogen, H_2S. It is known now that the mine gases are nearly all mixtures containing the said gases in varying amounts according to the many circumstances met with in mining. They are described by the miner as some form of "damp," *e.g. blackdamp* contains varying proportions of carbon dioxide; *firedamp* is made up largely of methane; *whitedamp* is the name given to small quantities of carbon monoxide; *stinkdamp* consists in part of sulphuretted hydrogen. The first two mentioned are the gases most frequently met with in mines, and generally blackdamp is the more frequent of these two.

Blackdamp, Chokedamp, or Stythe.—This gas was formerly thought to consist entirely of CO_2, or of a practically definite proportion of 13 per cent. CO_2 to 87 per cent. N_2, which idea is now known to be erroneous, the gas being of variable composition. Blackdamp is the name given by the miner to any atmosphere

* Dr Harger, "Explosions in Mines," vol. xliii. 133, *Trans. Inst. M.E.*
† *Queensland Govt. Mining Journal*, 15/8/12, p. 225.

which will extinguish an oil-lamp flame without exploding, or is non-injurious whilst a light will burn in it. The amount of CO_2 in any sample of blackdamp is very unstable, varying from a few per cent. up to over 20 per cent., and appears to depend chiefly on the source of its production.* It is considered to be a residual product of the oxidation of coal and other substances by air. It always contains a dominant proportion of nitrogen with the CO_2, and small amounts of oxygen and often carbon monoxide.

The Sources of the CO_2 in Blackdamp are— †

1. The breathing of men and animals and burning of lamps.
2. Small portions given off by coal during working.
3. The oxidation of coal and other carbonaceous materials (1 c.c. of coal will on the average absorb 2·1 c.c. of oxygen).
4. The oxidation and decay of timber.
5. The action of pit waters on carbonates.

The Sources of Nitrogen in the Blackdamp are—

1. The using up of the atmospheric oxygen by men, animals, and lights.
2. The gases given off by coal contain portions of nitrogen.
3. The using up of atmospheric oxygen by the oxidation of coal and other carbonaceous materials.
4. The using up of the oxygen by timber.
5. The oxidation of iron pyrites also causes the withdrawal of some of the oxygen of the air.

These factors enhance the percentage of nitrogen and carbon dioxide at the expense of the oxygen, and we may therefore expect to find blackdamp in old workings and anywhere where the ventilation is inadequate and also often in return airways. Blackdamp produced solely by the oxidation of coal might be expected to have the following approximate composition : —

Nitrogen . . .	94 to 95 per cent.
Carbon dioxide . .	5 per cent.
Carbon monoxide .	1 to 0·5 per cent.

Although as high as 31 per cent. of CO_2 to 69 per cent. of N_2 has been the composition of a blackdamp found issuing from the coal itself,‡ the usual limits are 5 to 20 per cent. of CO_2. Mr

* *Report* (Cd. 4551), 1909.
† J. Ivon Graham : vol. lv., *Trans. Inst. M.E.*
‡ Vol. xlviii., *Trans Inst. M.E.*

Graham, in addition to the above coal oxidation figures, considers that the blackdamp arising from rotting timber would yield a high proportion of CO_2, even up to 20 per cent., and that from the oxidation of coal containing iron pyrites (FeS_2) up to 15 per cent. The yield of firedamp greatly affects the composition of the blackdamp, especially the CO_2 content.

The effect of CO_2 on man is inappreciable if inhaled in low percentages, 2 per cent. only causing deeper breathing. It is not until 6 to 8 per cent. is reached that poisonous symptoms appear, and as much as 25 per cent. of CO_2 (not blackdamp) has to be present to cause danger to life. Lights are extinguished in atmospheres containing CO_2 as a result of the using up of the oxygen rather than the direct presence of CO_2. When the percentage of CO_2 is about $3\frac{1}{2}$, and the oxygen falls to about 17 per cent., the atmosphere is extinctive. About 15–20 per cent. of blackdamp will extinguish a flame, and 60 may be fatal, dependent on its composition. Pure blackdamp is irrespirable. A calculation of the constituents of blackdamp as given above, by the oxidation of coal, will give a sp. gr. very slightly greater than that of air, but increasing percentages of CO_2 will of course increase its density, which accounts for its being found most often on the floor of mines. The presence of firedamp decreases its sp. gr. by lowering the total amount of blackdamp present, though it often enriches the CO_2 content. Testing for blackdamp is a subject fraught with difficulties and almost impossible to accomplish accurately. Among the methods of testing in use mention may be made of the following :—

(*a*) Dr Haldane's Tube and Taper test.*

(*b*) Dr Haldane's Gas-Testing Apparatus (by diminution of volume on absorption).

(*c*) The Briggs Loop.

In any case the extinction of the candle or safety-lamp flame must be taken as a sure indication.

Hydrocarbon Gases are the important combinations of carbon and hydrogen which unite in groups or series, each series being typified by a similarity in the ratio of C and H entering into combination and by distinct characteristics. Thus the "paraffins" belong to the order of compounds expressible by the general formula C_nH_{2n+2}, the "olefines," C_nH_{2n}, the "acetylenes," C_nH_{2n-2}, the "benzenes," C_nH_{2n-6}, and so on. Each

* "Investigation of Mine Air," vol. xliii. p. 64, *Trans. Inst. M.E.*

member of a particular series is called a homologue, methane, CH_4, being the lightest of the paraffins, ethane, C_2H_6, propane, C_3H_8, butane, C_4H_{10}, etc., are known as the homologues of methane or the paraffin homologous series. This series is also called the fatty group, so distinguishing it from the olefine or oily group, and the benzene or aromatic group. There are eight hydrocarbon groups, but we are only concerned here with the methane series and chiefly with methane itself.

Firedamp, marsh gas, or mine gas, was formerly known also as light carburetted hydrogen, a term now abandoned owing to the confusing of it with another carbon-hydrogen compound somewhat similar. It is a mixture composed chiefly of methane, CH_4, frequently over 90 per cent., but it may be as low as 60 per cent., the remainder being nitrogen, hydrogen, carbon-dioxide, and oxygen in varying small proportions.

The following explanation of firedamp by a very eminent authority should be carefully read by every student:* "The firedamp of British mines contains as its inflammable constituent methane in predominating quantity. In only a few out of the multitude of samples analysed at the Experimental Station (Eskmeals) has any trace of ethane (C_2H_6) been detected, the highest percentage found being 2 (with 95 per cent. of methane). Carbon monoxide is usually present in the firedamp obtained from boreholes to the extent of 0·5 to 1·0 per cent., and occasionally (doubtful) traces of ethylene (C_2H_4) have been recorded. Hydrogen has never been found. These statements do not apply to such inflammable gases as may be present in limited quantity in the neighbourhood of a fire or heating in the coal. Near a heating, appreciable quantities of the higher members of the paraffin series of hydrocarbons (ethane, C_2H_6, propane, C_3H_8, and butane, C_4H_{10}) may be detected, as well as carbon monoxide. Their presence is, indeed, a sure indication of the occurrence of heating, antecedent to spontaneous combustion, in the coal. Near an active fire both hydrogen and carbon monoxide may be found. The uninflammable constituents of firedamp are nitrogen and carbon-dioxide, and the proportions in which these two are present vary considerably, dependent, no doubt, upon the proportion of blackdamp mixed with the firedamp. For all practical purposes, therefore, the firedamp of British coal mines may be regarded as methane."

The composition of the "blower" or continuous hissing flow

* Dr R. V. Wheeler on "Firedamp Explosions," *Trans. Inst. M.E.*, lx. 74.

which has existed on the surface at Porth, S.W., for over fifty years, is as follows :—

Methane . . .	95·80 per cent.
Carbon-dioxide . .	0·68 ,,
Oxygen . . .	0·25 ,,
Hydrogen . .	0·20 ,,
Nitrogen . . .	3·07 ,,
	100·00

This gas is now being supplied in cylinders to educational authorities. For all practical purposes we may consider firedamp to be methane. Methane, CH_4, the first member of the methane series of compounds, each member of which, or each homologue, has a chemical formula of the form of C_nH_{2n+2}, has a molecular weight of 16 and therefore a sp. gr. of $\frac{16}{14{\cdot}4 \times 2} = 0{\cdot}559$. It can be prepared in several ways, of which the following are instances :—

(*a*) By heating sodium acetate, $CH_3{\cdot}COONa$, with caustic soda, NaOH, in an iron tube to a high temperature, the following reaction takes place :—

$$CH_3{\cdot}COONa + NaOH = CH_4 + Na_2CO_3.$$

The methane produced is not of much use in mine experiments as it contains impurities often up to 3 per cent. and has a luminous bluish flame, whilst the flame of pure methane is pale blue and not very luminous.

(*b*) The action of water on aluminium carbide, Al_4C_3, when heat is applied, yields methane which can be easily purified by passing it over hot copper oxide.

$$Al_4C_3 + 12H_2O = 3CH_4 + 4Al(OH)_3.$$

(*c*) Methane chiefly results from the decay of vegetable matter in the presence of cellulose (usually indicated as $(C_6H_{10}O_5)_n$, the subscript n being somewhat doubtful). Cellulose is the chief component of woody fibre from which coal is derived in the processes of fermentation and mineralisation by nature. These processes are due to the combined action of bacteria, temperature, pressure, the gas-sealing action of water, and many other causes. Probably the result may be represented thus— *

$$2C_6H_{10}O_5 = 5CH_4 + 5CO_2 + 2C,$$

* See Lewes, *The Carbonisation of Coal*, chap. i.

which most likely is the origin of methane in coal. It is certainly liberated in areas where vegetable matter ferments and decays, hence the name marsh gas. It is very inflammable, being explosive between certain limiting percentages in air, and when burnt in oxygen there is a diminution in bulk equal to its own volume, thus—

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

1 vol. -2 vols. $=$ 1 vol. (water volume negligible).

The volume of CO_2 formed equals that of CH_4 burnt, a fact made use of in Haldane's * and other gas-testing apparatus where CH_4 is ignited in the presence of O. Methane is a light, colourless, odourless, and tasteless gas, burning with a pale blue flame in air and is usually found near the roof in mines. It is very inflammable, though it will not support combustion alone. (By inflammation is meant the evolution of light and heat in the vicinity of the source of ignition.† By explosion is meant the continued propagation of the flame throughout the mass.‡)

When mixed with air, methane gives a characteristic cap which exists as an envelope, long before it can be seen, for small percentages on an oil-lamp flame. (*Note.*—When speaking of firedamp or methane ignitions and explosions a mixture of methane and air is understood, since methane is not explosive alone.) Representative caps give fairly faithful records of the firedamp present up to about 5 per cent., when further addition causes the flame to traverse the mixture even after the light is put out, and if the lamp is retained in it, alight, it ultimately extinguishes the lamp light. This cap or halo is due to sufficient heat being present near the flame to ignite the methane (650–750° C.).§ The cap increases in luminosity and dimensions as the percentage of methane increases. The luminant itself often provides a fuel cap which can be recognised in clear atmospheres with a low flame, and of course the width of the wick will affect the height of the cap in the approximate ratio as in fig. 33, where *b* is the breadth and *h* the height. (*Note.*—The cap is not merely an extension of the test flame but is an envelope covering it down to its base, whence it should be measured for height.)

* See Dr Haldane's *Methods of Air Analysis.*

† See Dr R. V. Wheeler on "The Ignition of Firedamp," *Trans. Inst. M.E.*, lxiii.

‡ See Dr R. V. Wheeler on "Firedamp Explosions," *Trans. Inst. M.E.*, lx. These two splendid papers must be read by all interested in this aspect of mining.

§ Dixon V. Coward, *Journal of the Chem. Soc.*, xcv.

The cap is extremely indistinct, and partially formed at 1 per cent.; only an expert observer can detect such quantities.

At $1\frac{1}{2}$ per cent. the tip of the cap is still unrecognisable but the cap itself is bigger.

At 2 per cent. the tip of the cap is just visible, the whole cap being almost an equilateral triangle.

At $2\frac{1}{2}$ per cent. the luminosity has increased and the cap is much more intense and slightly higher, the height being equal to the base.

At 3 per cent. the cap increases in height by a greater

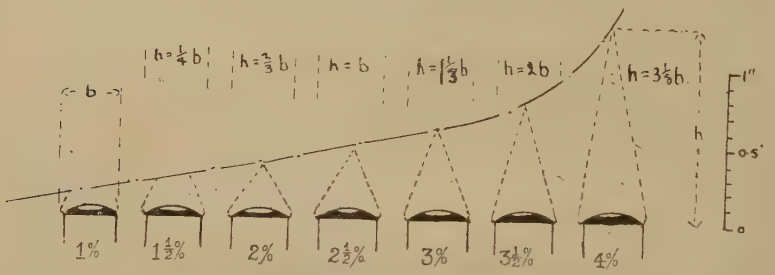

FIG. 33.—Mine gas caps, with safety lamp burning a mixture of colza and paraffin oils.

increment than previously, where it was nearly uniform, and is now $1\frac{1}{3}$ times as high as it is wide.

At $3\frac{1}{2}$ per cent. the elongation is more pronounced, the height being twice the base, and its outlines are very well defined.

At 4 per cent. the cap is beginning the extreme attenuation which marks the limit of practical accuracy.

At $4\frac{1}{2}$ per cent. the top of the cap is unstable and well inside the gauze; spiring and bellying of the cap commences.

At 5 per cent. the top of the cap is no longer a point but rounded, and is sometimes wider than the base, the cap being contracted to a waist in the mid-height, and extinction soon follows in an oil lamp.

Many devices are in vogue for intensifying the luminosity of the cap, so enabling a small percentage to be more easily recognised. Among these may be mentioned—

(*a*) **The Cunningham-Cadman Device,*** which consists of a small plate which can be swung into the lamp flame by a pricker arrangement. The plate contains chemical salts like carbonate

* *Trans. Inst. M.E.*, xxxix. 13; *Ibid.* xliii. 68.

of soda and cuprous chloride, which impart a distinctive green coloration to the otherwise yellow flame. Thus, if the plate is inserted into the cap zone the two flames can be easily distinguished (see fig. 34).

(*b*) **The Briggs Loop,** introduced by Professor Briggs of Edinburgh, consists of a small 22 S.W.G. copper wire which, being a good conductor of heat, abstracts heat from the flame in sufficient quantities to reduce the luminosity of the oil lamp test flame. By this means a larger test flame can be used. If the loop (see fig. 35) be dipped in zinc-chloride a green coloration is given to the flame which is, however, temporary.

(*c*) **Clowes' Hydrogen Lamp.**—This is a Hepplewhite-Gray lamp,

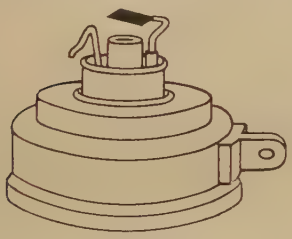

Fig. 34.

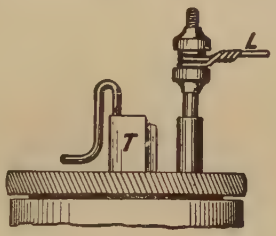

Fig. 35.

modified to burn hydrogen from a cylinder which is attached when reading to test for firedamp. The hydrogen flame is intensely hot, and for this reason any methane present gives a bigger cap, which is read off on a special scale within the lamp. This lamp soon gets misty, due to the formation of steam by the burning hydrogen, and this affects the accuracy of the readings, as also does the pressure and quality of the hydrogen. Furthermore, there is the danger of a hydrogen explosion in the lamp, which causes it to be debarred from mines where it would be most useful.

(*d*) **Stokes' Alcohol Lamp** is also a modified Hepplewhite-Gray lamp, and depends for gas-testing on the principle that alcohol burns with a blue non-luminous flame, so showing a more visible cap. Ordinarily the lamp burns oil from a separate compartment, and the alcohol is resorted to when a cap is invisible or partially visible with the oil flame.

(*e*) **Beard-Mackie Detector.**—We have seen that a cap exists for small percentages of methane before it is visible owing to the masking effect of the luminous flame of the lamp. In the

Beard-Mackie lamp (fig. 36) the heat generated by the invisible small cap heats up the bottom of a series of platinum rings on a fixed stage, causing them to glow. The number of rings glowing is the measure of the percentage of gas present.

Among the many inventions for detecting and estimating the presence of methane with the electric lamp the principle of the heating effect upon a tungsten or platinum wire, first employed by Liveing, will be found to be fairly general. Among the more successful methods of detecting firedamp (in large and small quantities) mention should be made of the device * used by T. J. Thomas (Cwmmer Colliery), which is as follows :—

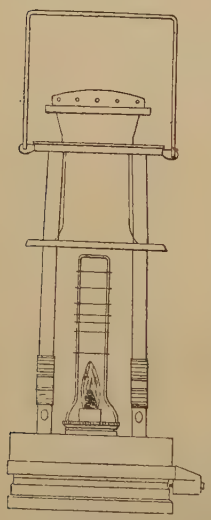

Fig. 36.

The diagrams 37 and 38 show the manner of applying the platinum wire in connection with the miner's electric lamp; a = battery (cell); b = lamp; c = short length of platinum wire; d = resistance in parallel with the platinum wire. The resistance of the wire c is such that the current passing to the lamp will cause the platinum wire (even after the cell has been run for nine hours) to be heated to a degree sufficient to react with firedamp. Varying percentages will cause an observable variation in the temperature of the platinum wire and a reduction in light of the lamp. The initial temperature of the wire can be fixed so as to cause the wire to fuse in the minimum explosive mixture of firedamp and air; in a percentage of methane which would be dangerous to life, or a mixture of this kind which is explosive (containing 34 per cent. of blackdamp), viz. firedamp, 6·25 per cent.; blackdamp, 34·16 per cent.; air, 59·59 per cent.

Fig. 38 shows an arrangement whereby the platinum wire would fuse in a dangerous mixture of firedamp and air, and, after fusing, allow sufficient current to pass through the resistance, which is in parallel, to enable the lamp to give sufficient light for the user to withdraw from the place, having given him an unmistakeable indication that firedamp was present. The intensity of brilliancy of the platinum wire in various percentages

* *Col. Guardian*, 14/2/19.

of firedamp and air can be compared with the lamp filament, which can be used as a contrast. The lamp, therefore, provides a means of indicating percentages of methane from 0·5 per cent. upwards.

The number of devices and inventions for the detection and estimation of firedamp are far too many to describe or enumerate here. We have just dealt with a few practical applications of different principles, and the reader is now referred elsewhere.* The effect of the presence of blackdamp is to damp or lessen the luminosity and size of firedamp caps, while the volume of the feed also affects the cap directly.

Coal gas as supplied for domestic purposes contains a large percentage (up to 55 per cent.) of hydrogen, consequently the percentage of methane is only about 30–35 per cent. The result is that in comparative tests on town-main coal gas and methane the former gives a lower cap than the latter for the same percentages.

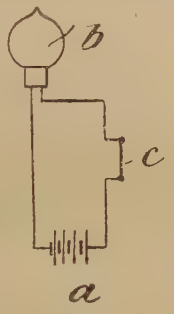

Fig. 37.

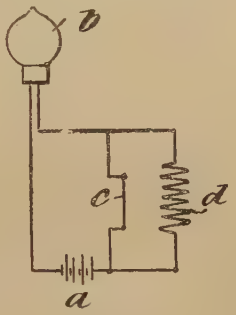

Fig. 38.

When speaking of percentages, it will be understood that we mean percentages by volume, which, presumably, is meant where similar percentages are mentioned in the C.M.A., 1911, and regulations. At least by adopting a percentage by volume interpretation we are on the safe side, *e.g.* $2\frac{1}{2}$ per cent. methane by volume is not $1\frac{1}{2}$ per cent. by weight,

$$i.e.\quad \left.\begin{array}{l} 2\frac{1}{2}\times 8 = 20 \\ 97\frac{1}{2}\times 14{\cdot}4 = 1400 \end{array}\right\} = \frac{20}{1400}\times\frac{100}{1} = 1\frac{3}{7} \text{ per cent. by weight.}$$

Methane, apart from its inflammability, has fairly well defined explosive limits.

5·6 per cent. CH_4 in an air-methane mixture = lower explosive limit.

14–15 per cent. CH_4 in an air-methane mixture = higher explosive limit.

* Foster and Haldane, *Investigation of Mine Air*; Turquand, "Methanometers," etc., vol. xlviii., *Trans. Inst. M.E.*

9·4 per cent. of methane to 90·6 of air is the most violent explosive proportion.

At about 15 per cent. the spontaneous passage of flame through the mixture ceases, and at 25 per cent. the flame is extinguished.

In certain kinds of firedamp the limits of explosibility vary somewhat from the methane figures above. This is because they contain heavy hydrocarbons (especially firedamp from old goafs), and the heavier the hydrocarbon the wider its explosive limits, the less it takes to make an explosive mixture, and the more easily is the mixture ignited.

For coal-gas the lower and higher limits are 5 per cent. and about 28 per cent. respectively.

In the above most explosive proportion (9·4 per cent.) the gas burns according to the following reaction :—

$$CH_4 + 2O_2 = CO_2 + 2H_2O.$$

If the air or oxygen supply is limited CO_2 is not formed, but CO, thus—

$$2\,CH_4 + 3O_2 = 2CO + 4H_2O.$$

The steam condenses, and there is a contraction of 2 volumes out of 3 in the first case, and from 5 volumes to 2 in the second case, which are phenomena of all gas explosions in mines. The calorific power, *i.e.* the heating value, of methane is 11,850, which means that 1 lb. of methane in combination with oxygen gives sufficient heat to warm 11,850 lbs. of water up from 0° to 1° C. If one calculates the calorific intensity of a $2\frac{1}{2}$ per cent. methane-in-air mixture one finds the result is a rise of 650° C. above normal (say 680° C. in all), and as the ignition temperature of methane-in-air mixtures is 650–750° C. an explosion is therefore possible, theoretically, at $2\frac{1}{2}$ per cent.

Pressure and temperature conditions may vary over a rather wide range without affecting the explosibility of methane-air mixtures.* Initial temperatures of up to 800° C. only reduce the lower limit by 1 to $1\frac{1}{2}$ per cent., while pressures of up to 5 atmospheres have no effect upon it, so that the deepest mines would show no change in the explosive limits from this cause. The nature, method, and source of ignition, also the shape of the vessel for ignition, *will* have effect on the range of explosibility.

* *Bureau of Mines, U.S.A., Technical Paper* 121.

TABLE VII.—*Table of Ranges of Explosibility of Mixtures of well-known Inflammable Gases and Air.*

Gas.	Symbol.	Lower limit, per cent.	Higher limit, per cent.
Methane . .	CH_4	5·6	14·8
Hydrogen . .	H_2	5	72
Carbon monoxide .	CO	13	75
Ethane . . .	C_2H_2	4	22
Coal gas . .	. .	5	28

Assuming the composition of air to be 21 per cent. oxygen and 79 per cent. nitrogen, a mixture of methane and air in the most explosive proportions (*i.e.* 9·4 of methane to 90·6 of air in every 100 parts) will contain—

$$21 \times \frac{90{\cdot}6}{100} = 19{\cdot}026 \text{ per cent. of oxygen}$$

$$79 \times \frac{90{\cdot}6}{100} = 71{\cdot}574 \quad \text{,, of nitrogen}$$

$$9{\cdot}4 \quad \text{,, of methane}$$

$$\text{Total } 100{\cdot}000$$

and similarly the total quantities may be deduced for any existing percentages of methane and air.

The atmosphere may suffer a considerable reduction in the oxygen content before methane, if in suitable proportion, is rendered incapable of explosion, as the following findings of the Explosions in Mines Committee show :—*

TABLE VIII.

Per cent. oxygen in the atmosphere.	Lower limit of explosibility ; per cent. of methane.	Upper limit of explosibility ; per cent. of methane.
20·9	5·6	14·8
19·4	5·8	12·9
17·0	5·8	10·5
15·8	5·9	8·9
14·8	6·1	8·3
13·8	6·4	7·3
13·5	6·5	6·8
13·2 and under, no	mixture is capable of	propagating flame.

* *Sixth Report*, Explosions in Mines Committee, 1914.

It will be noted that a gradual restriction of the explosive limits accompany the diminution of oxygen.

Outbursts of Gas.—An outburst of gas contrasts very markedly with a blower, in that the latter is a more or less steady outpouring of gas from the seam, roof, or floor, accompanied usually by a hissing or droning sound, whilst an outburst is a sudden release of gases under great pressure, often over 600 lbs. per square inch, and frequently moving large quantities of coal, etc., in escaping. Folding, reversals, and high inclination of the strata appear to be the factors influencing surface and borehole outbursts, but in the mine other additional factors, which are only vaguely comprehended, assist in producing the outburst. Although outbursts are not common in this country the liability exists and is serious, especially when we think of a certain instance in one of the highly contorted Belgian seams where 12 million cubic feet of firedamp were suddenly liberated, causing a death-roll of over 120. The outburst is usually inflammable gas, but instances are on record of large bodies of CO_2 being released suddenly.*

One of the additional causes of outbursts in mines is that of "weighting," due to long coal faces and scanty packing, and among the symptoms in such cases, if the outburst is not in the solid, will often be found blisters or bulges in the roof and floor which increase and burst. These may be tapped by boreholes or pricked in the early stages. It should be understood that a cavity or reservoir of gas is not implied here as the source of the gas, but rather the multitude of interstitial spaces in the coal or strata. This is evidenced by the type of coal and dirt which usually accompany these outbursts, it being fine, very dry, slickensided, and extremely friable.

The Ponthenry outburst† was accompanied by very fine coal altered, as a result of geological disturbance, in the immediate locality, into a friable mass yielding only pea sizes. Obviously this with hard floor and roof is eminently suitable for the storage of gas, which in this case contained 92 per cent. of firedamp, and was, as in nearly all such cases, perfectly dry.

The Tareny outburst occurred under somewhat similar conditions to the above, the out-blown coal being anthracite, which yielded, on heat treatment, only 54 c.c. instead of about 500–600 c.c. of occluded gases per 100 grams, as is usual with anthracite. This shows the source of the body of gas which burst out. (A

* Prof. Brigg, *Trans. Inst. M.E.*, 1921.

† Geo. Roblings, *Trans. South Wales Inst. Eng.*, xxxvi. 423.

gas is said to be occluded when it is taken in and pent up in the pores of a substance.) The gas in the Tareny instance contained large quantities of CO_2 and much heat was experienced after the outburst, which is contrary from what is to be expected on sudden expansions. This probably was due to oxidation, which can be put down as another addition to the features of outbursts. Professor Briggs attributes the causes of sudden outbursts of gas to four factors existing together, viz.—

(*a*) The presence of gas under pressure.

(*b*) The presence of a mass of disintegrated coal loose enough to move when pressure is suddenly relieved, and which sets free its own absorbed gas.

(*c*) Absence of water.

(*d*) Geological and other conditions preventing the normal slow escape of gas from the coal.

Carbon Monoxide, CO, or Whitedamp.—This dangerous gas, also known as carbonic oxide, is produced normally in most British mines, but in such small quantities that its physiological effects may be considered negligible. Its density is about 14, whence its sp. gr. is 0·972. It is colourless, odourless, tasteless, and burns with a blue flame. It is produced in quantities as a result of the incomplete combustion of carbon or carbonaceous matter in oxygen, and therefore is found after colliery explosions, shots, and in coke stoves. It forms the blue flames which are often seen on domestic fires. In producer-gas plants, coke ovens and retorts, and anywhere where the oxygen supply for burning is limited, it is found in large quantities. It may be prepared by passing CO_2 over hot charcoal, when the following reaction results :—

$$2CO_2 = O_2 + 2CO.$$

It is also easily prepared by the action of sulphuric acid on formic or oxalic acid, the gas being collected by upward displacement over water, since it is not very soluble in water. It is explosive in certain proportions with air, as already stated, and is extremely poisonous. The red colouring matter of the blood, or hæmoglobin, which distributes the inhaled oxygen of the lungs to the tissues, has between two hundred and three hundred times greater an affinity for carbon monoxide than for oxygen. Normally the hæmoglobin combines loosely with the oxygen and gives it up in its circuit of the human system by a complex oxidising reaction in the tissues, so accumulating carbonic acid

gas, *i.e.* becoming carboxyhæmoglobin from oxyhæmoglobin. The effect of CO is to ruin this function of the red corpuscles, causing death very quickly when inhaled, even in such low proportions as less than 1 per cent. The effect also is not local—many vital organs will be affected—nor is it momentary but cumulative. Toxic symptoms appear in different persons at different times, dependent chiefly on the state of health and physical exertion of the individual, the length of time of exposure to the gas, and the percentage of oxygen in the atmosphere. Dr Haldane * states that while only 0·2 per cent. would be very dangerous, even as low as 0·02 per cent. will, after sufficient time, say about twenty minutes, cause headache and partial disablement. The above authority further states that 0·1 per cent. of CO produces the same degree of blood saturation as the normal oxygen content of the air owing to the much greater affinity of the blood for CO. The saturation percentage increases with the exertion, as also does the cumulative effect mentioned above. This points to 0·1 per cent. as being dangerous to life. The effect on mice and birds, especially canaries, is more marked than upon man, the birds being distressed after about one to two minutes' exposure to an atmosphere containing 0·2 per cent., and later falling insensible. The effect upon man is somewhat more sluggish but is none the less certain.†

It may be detected and estimated by the following methods :—

(*a*) The "blood test."
(*b*) The Graham apparatus.

(*a*) The blood test consists of saturating the blood of a mouse with CO and diluting it with water. A solution of blood from a finger-prick is now saturated to the same extent. The tint of the two solutions are compared as a guide to the quantity of CO present, but for more accurate work titration with carmine solution is resorted to. This method is not recommended for mine work owing to the poor light for colour work, the time and care needed, and the type of apparatus, which make it a matter of much expertness for the investigator.

(*b*) **The Graham Apparatus.**‡—We have seen that CO is produced when blackdamp is produced as a result of the oxidation of coal, but the amounts are so small as to need special methods

* *Methods of Air Analysis.*
† See J. W. Whitaker, *Physics and Chemistry of Mining.*
‡ See *Colliery Guardian*, 25/4/19.

for their estimation. The Graham apparatus is portable and applies the principle of the oxidation of CO by iodine pentoxide, I_2O_5, at a temperature of between 80° and 140° C., according to the following equation :—

$$5CO + I_2O_5 = 5CO_2 + I_2.$$

A known volume of the air to be analysed is passed over the iodine pentoxide tube contained in oil in an electrically-heated

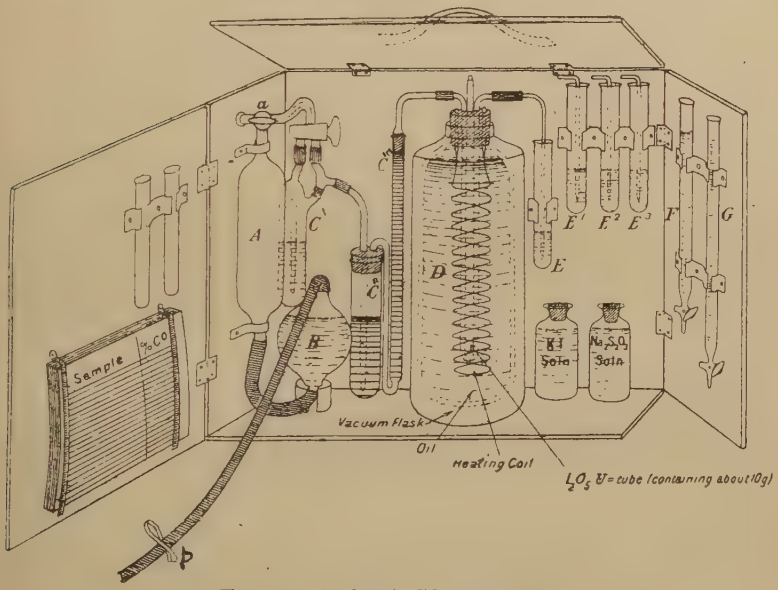

Fig. 39.—Graham's CO apparatus.

tube (after removal of all the gases which interfere with the reaction). By this means the CO is oxidised to CO_2, and iodine liberated in a corresponding proportion. This iodine is absorbed in a solution of potassium or sodium iodide (10 per cent.) and titrated with a weak standard solution $\left(\text{about } \frac{N}{500}\right)$ of sodium thiosulphate, using starch solution as an indicator. The apparatus gives an accuracy of 0·005 per cent., and a complete test is carried out in about five minutes.

The apparatus, fig. 39, consists of a gas-sampling vessel, A, of

known volume, having a three-way cock *a*. B is a reservoir connected to A to enable a sample to be passed in at *a* by displacement of the water in it. A set of absorption tubes, C′, C″, and C‴, contain in C′ bromine in potassium bromide solution, in C″ saturated caustic soda solution, and finally in C‴ solid caustic soda, calcium chloride, and phosphorus pentoxide respectively for purifying and drying the gas sample. The vacuum flask, D, contains oil, and holds, from the bung, a U-tube carrying about 10 grams of I_2O_5, a thermometer, and a contact maker for the electric heating off the lighting circuit. The delivery tube E is rubber connected to the tube of iodine pentoxide at one end and at the other dips into potassium-iodide solution in a test tube. The gas from the apparatus comes through at E in small bubbles, thus completely absorbing the iodine. E^1, E^2, etc., are spare test tubes containing 10 per cent. solution of potassium iodide. F and G are graduated burettes containing 1 per cent. starch solution and sodium thiosulphate of such strength that two drops in the titration equal 0·01 per cent. of CO when using a 100 c.c. sample.

In taking a sample, A is filled by blowing through the pinch clip *p* the water from B, *a* being open. Then the pressure in B is released and the water returns to B and a sample passes in at A to a desired mark, say 100 c.c. Now shut off the three-way tap, *a*, from A and the atmosphere, open it to the purifiers and driers C′, C″, etc., and pass the sample into the apparatus; take in another sample (as the first is a "washing out" process with E disconnected). Join up E, pass the sample through the apparatus, then disconnect E from its clip. E is rinsed with the potassium-iodide solution it contains, and a few drops of starch solution are added, which, if CO be present, gives a bluish-violet colour. Standard thiosulphate solution is then used drop by drop as an indicator until all the coloration has disappeared. The apparatus weighs only 6 lbs.

The findings of Dr Haldane are briefly as follows: -

(*a*) The physiological effects of CO normally produced in mines are negligible.

(*b*) Small animals, though more susceptible to poisoning by CO than man, recover more quickly from its effects.

(*c*) Addition of CO_2 palliates, whilst depletion of oxygen accentuates the poisonous effects of CO.

(*d*) In ordinary air containing less than 0·02 per cent. CO no noteworthy toxic symptoms occur.

(*e*) Afterdamp contains large and variable quantities of CO.

The findings of Mr Graham are briefly as follows : —

(*a*) Carbon monoxide is found normally in the return air of pits over 500 yards deep.

(*b*) It is produced as a result of the oxidation of carbonaceous matter.

(*c*) The existence of old workings, unworked pillars, etc., are likely to increase its production.

(*d*) It is a good guide to the investigation of heatings, but it cannot be detected with the safety lamp.

The findings of the Medical Profession are briefly as follows :— *

(*a*) Carbon monoxide is innocuous up to a small percentage in air of coal mines.

(*b*) It is easily dissociated from hæmoglobin.

(*c*) It may be found by spectroscopic examination in blood.

(*d*) It can act detrimentally on nerve tissue.

(*e*) It does not damage lung tissue.

(*f*) It can cause anoxæmia (tissue starvation) and this in turn may cause nystagmus.

(*g*) Hæmoglobin has about 240 times greater an affinity for carbon monoxide than for oxygen.

Sulphuretted Hydrogen, H_2S, or Stinkdamp.—This gas, also known as sulphide of hydrogen, is not of very frequent occurrence in mines, and, when present, is due to the action of minute organisms or bacteria on animal and vegetable matter, *i.e.* timber, etc. Mr Winmill (Doncaster) has found quantities of gelatinous matter in certain mine workings which are organic, living on calcium sulphate and evolving H_2S. The action of acid waters on the sulphur found in coal also produces H_2S. This sulphur occurs in coal as FeS_2 in the cubic crystal as iron pyrites, and the rhombic crystal as marcasite. It also occurs as sulphate of lime, $CaSO_4$, as ferrous sulphate, $FeSO_4$, and in small percentages as free sulphur. A frequent indication of H_2S in the mine arises when heatings occur previous to outbreaks of gob fires, its smell being considered in many cases a distinct indication of an incipient gob fire. It is produced in the manufacture of town gas, the gas being purified from it by passing it over ferric oxide, Fe_2O_3. Some firedamps produce it, also certain colliery waters from old wastes. It may be prepared by the

* Frederick Robson, M.D., Medical Symposium on "Coal Miners' Nystagmus," before S.W. Inst. Engrs., 22/3/23.

action of weak cold sulphuric acid on ferrous sulphide, FeS, in a fume chamber, producing ferrous sulphate, thus—

$$FeS + H_2SO_4 = FeSO_4 + H_2S.$$

Its presence can be very surely detected by holding a piece of white paper moistened with lead acetate solution in the atmosphere where the gas is suspected. The paper will blacken if H_2S be present, due to the formation of sulphide of lead, PbS.

Stinkdamp, as its name implies, has a vile smell, which is usually described as resembling that of rotten eggs. It has a sweetish taste, is not a supporter of combustion, though it will burn when the proper proportion of oxygen or air is present, and is very soluble in water. Water at N.T.P. will dissolve $4\frac{1}{2}$ to 5 times its own volume of H_2S. Like carbon monoxide it is extremely poisonous, from 0·2 to 0·5 per cent. soon causing death, the effect being, as in the case of carbon monoxide, more rapid on birds and mice than on man. Although the sense of smell is paramount in detecting mine gases it should always be supported by chemical and other means, because in odoriferous atmospheres the sense soon becomes adapted or numbed. The gas will be seen to have a density of 17, *i.e.* it is 1·18 times as heavy as air, and will, therefore, accumulate near the floor, where it will be detected by its smell and by the fact that the flame of an oil lamp will gradually lessen in it.

Table IX. gives a convenient summary of the principal physical properties of the " mine gases."

The Effect of Mine Atmospheres on the Mine.—Solid coal is extremely air-tight even under pressure; CO_2 passes through coal much slower than CH_4, which in turn is slower than H. The quantity of oxygen that can permeate any distance in coal is negligible.* Fires inside pillars are due to breaks in the coal and not to permeability. Since solid coal is practically impermeable to air, changes of barometrical pressure will have no appreciable influence on emissions of firedamp from the solid; barometric change effects in these cases apply to gases issuing from fissures, etc. Now if the coal be impermeable we can understand the accumulation of pressure previous to an outburst of gas.

Solid Solution or Adsorption.—The process of dissolving or taking up gases is altogether different from permeability or compression: solid coal is capable of dissolving considerable quantities of different gases. The action, a physical one, is

* J. Ivon Graham, *Trans. Inst. M.E.*, lii. 340.

TABLE IX.—*Physical Properties of the Mine Gases.*

	Air.	Oxygen.	Hydrogen.	Nitrogen.	Carbon Monoxide.	Carbon Dioxide.	Methane.	Hydrogen Sulphide.	Remarks.
Chemical symbol	..	O_2	H_2	N_2	CO	CO_2	CH_4	H_2S	
Atomic weight	..	16	1	14	..	..	..	..	
Molecular weight	..	32	2	28	28	44	16	34	Recalculated to nearest figure with O=16.
Density (grams per litre)	1·293	1·429	0·0899	1·25	1·25	1·976	0·717	1·539	(Rayleigh) 0° C. and 760 mm.
Relative Density (H=1)	14·4	16	1	14	14	22	8	17	To nearest figure at N.T.P.
,, ,, (air=1)	1	1·11	0·0695	0·97	0·972	1·53	0·559	1·18	
Melting-point	..	−219°	−259°	−210°	−207°	−65°	−184°	−86°	Centigrade.
Boiling-point (one atmosphere press.)	−192°	−183°	−253°	−196°	−190°	−78°	−164°	−62°	Centigrade.
Solubility in water	2·918	4·9	2·15	2·4	3·5	171·3	5·5	468	Volumes dissolved in 100 vols. of water at N.T.P.
Corresponding critical data: Critical temperature	−140°	−118°	−234°	−146°	−141°	31°	−95·5°	100°	Centigrade.
Corresponding critical data: ,, pressure	39	50	20	33	36	73	50	88·7	Atmospheres.
Corresponding critical data: ,, volume	·00517	·00426	·00264	·00517	·0050	·0066	·00488	·00578	Ratio of vol. at crit. T. and P. to that at N.T.P.
Heat of formation	..	..	..	..	29,000	94,300	21,700	2700	Calories per gram molecule.
Heat of combustion	..	..	68,000	..	68,000	..	212,000	133,460	Calories per gram molecule.
Specific Heat (*a*)	·2389	·242	3·40	·235	·242	·202	·591	·245	At constant pressure.
Specific heat (*b*)	·1715	·154	2·40	·175	·171	·165	·452	·183	At constant volume (equal weights at 20° C.).
Thermal conductivity	$5·22\times10^{-5}$	$5·63\times10^{-5}$	$31·8\times10^{-5}$	$5·24\times10^{-5}$	$4·99\times10^{-5}$	$3·2\times10^{-5}$	$6·47\times10^{-5}$	..	At 0° C. and sp. ht. at constant volume.
Adiabatic compression or expansion index (*n*)	1·401	1·400	1·419	1·410	1·401	1·300	1·313	1·340	Ratio of sp. ht. at constant pressure to that at constant volume.

termed adsorption, and is exclusive of any chemical action; the amount of gas taken up by adsorption can be recovered again by physical means. Adsorption is interpreted as an internal and external surface action—the gas is liquefied on the internal surfaces of the coal—therefore it will be necessary to consider it as a distinctly different phenomenon from simple compression of the gas in the coal pores. Mr Graham—quoted below—has shown that coal adsorbs methane and that the amount adsorbed increases with the fineness of the coal but decreases with the temperature, being, at ordinary temperatures, about three times the amount taken up at 100° C. Generally speaking, one gram of coal will take in this way about 2·4 grams of methane at ordinary temperatures, and the lower the percentage of methane in mine gas the higher the amount dissolved. CO is less soluble than CH_4; N_2 less still, and H_2 least of all, while CO_2 is much more soluble than CH_4.

Absorption.—This is the term applied to the taking up of gases by solids when the operation is accompanied by a definite chemical action. It is impossible to extract the oxygen or other gases so absorbed by any physical means because it is taken up by the coal substance to form an oxidation product. Messrs T. F. Winmill and J. Ivon Graham,* in the extensive and valuable researches noted below, have established from much experimental investigation certain facts respecting the taking up of gases, etc., by coal. These have such an important bearing on mine ventilation that we must, in order to comprehend subsequent chapters fully, include some of their findings here.†

The Absorption of Oxygen, etc., by Coal.‡—1. The amount of the oxygen absorbed is roughly proportionate to the carbonaceous content of the coal. It is also proportional, but not directly so, to the extent of the coal exposed and also the amount of oxygen in the air. It varies even up to about 8 c.c. per gram for various coals, soft coals absorbing the most.

2. The rate of oxidation is not proportional to the partial pressure of the oxygen but to the square root of the percentage of oxygen present in the air. Mr Winmill gives the following approximate formula where fairly arbitrary units are implied:—

$$A = K\sqrt{P} \quad (72).$$

A = rate of absorption.
P = percentage of oxygen.
K = constant, about 56.

* Messrs Winmill, Graham, and Hill, *Trans. Inst. M.E.*, vols. xlvi. to lii.
† *Trans. Inst. M.E.*, vols. xlvi. to lii.
‡ *Ibid.*, li. 498.

3. The *temperature* rises with the oxidation.

4. The absorption is continuous, increasing indefinitely with the time.

5. The *heat evolved* during the process of absorption decreases as the quantity absorbed increases; coals which absorb much oxygen give out less heat in proportion than coals which absorb little, also the more readily the coal absorbs oxygen the less is the heat evolved.*

6. As the *temperature* rises coal will absorb oxygen faster and the total amount absorbed is greater.

7. A considerable proportion of water is produced relative to the quantity absorbed.

8. At temperatures below 50° C. coal dust when moist will absorb oxygen at a rate approximately half as great again as when dry.

9. The absorption of oxygen by coal is independent of bacterial action.

Regarding **adsorption** of oxygen and other gases by coal the following data are presented:—

1. The presence of water diminishes the adsorptive capacity of coal.

2. The adsorption increases with the pressure of the gas.

3. During the adsorption of CH_4 by coal heat is evolved, and the converse occurs when the adsorbed gas is released. This latter means a cooling effect, and is especially noticeable in a gassy longwall working face where a considerable amount of CH_4 is drawing through breaks and fissures behind the coal face.† The heat absorbed when adsorbed gas is liberated is calculated to be twenty-eight large calories per cubic foot of gas given off, *i.e.* 110 B.Th.U. This accounts for the coal being often lower in temperature than the surrounding strata, for coal is a bad conductor of heat, and therefore does not readily transmit heat to balance the loss.

The Effect of Mine Atmospheres on Iron Pyrites.—The initial heating due to the oxidation of iron pyrites, FeS_2, or "brasses," is a contributory cause to mine heating, but from the preceding remarks it will be seen to be not the only or most important cause, as was formerly thought. The oxidation of coal itself is more extensive, and therefore more effectual, in causing heat in mines than that of iron pyrites. Highly crystalline pyrites are not very liable to oxidation, but friable pyrites or inferior

* *Trans. Inst. M.E.*, xlviii. 511. † *Ibid.*, 1921.

coal with pyrites soon disintegrate under the action of mine atmospheres, and in such cases the pyrites are very probably the determining factor in the oxidation. The effect may be expressed thus (in all equations showing oxidation by air only the oxygen component of the atmospheric mixture is considered)—

$$2FeS_2 + 7O_2 + 2H_2O = 2H_2SO_4 + 2FeSO_4.$$

(Pyrites) + (Oxygen) + (Water) = (Sulphuric acid) + (Ferrous sulphate)

To a less extent the reaction may be *

$$4FeS_2 + 15O_2 + 2H_2O = 2H_2SO_4 + 2Fe_2(SO_4)_3 \text{ (Ferric sulphate)},$$

and at red heat the reaction is totally different, thus—

$$2FeS_2 + 11O = Fe_2O_3 + 4SO_2$$

(Ferric oxide) + (Sulphur dioxide).

Mr Winmill finds that under ordinary mine conditions the oxidation of pyrites produces heat at the rate of 4·3 calories per c.c. of oxygen absorbed, or twice that produced by the absorption of an equal quantity of oxygen by coal. But as the sp. ht. of pyrites is only two-thirds that of coal the absorption of the same quantities of oxygen by iron pyrites and coal will raise the temperature of the pyrites three times as much as it would an equal weight of coal. The rate of heating due to the oxidation of iron pyrites varies greatly in different localities—the pyrites of Staffordshire, for example, having a much faster heating rate than those of Yorkshire. Generally speaking, the rate of oxidation heating by pyrites is several times as speedy as that due to the oxidation of coal, which fact is due to the low sp. ht. of pyrites and the high increase of oxidation in pyrites with rise of temperature.

The elaborate work † of Mr F. E. E. Lamplough and Miss A. M. Hill at the University, Cambridge, yields the following important facts regarding atmospheric oxidation :—

1. Oxidation of coal is increased if iron pyrites be present, evolving considerable heat.

2. Oxidation of coal dust takes place with gas containing much less oxygen than is present in normal air.

3. The heat produced is due to the oxidation (*a*) of coal dust or carbonaceous matter, and (*b*) of iron pyrites.

* See T. F. Winmill, "Oxidation of Pyrites," *Trans. Inst. M.E.*, vol. li.

† *Trans. Inst. M.E.*, xlv. 628.

4. The heat evolved when coal dust absorbs oxygen is about 3·3 calories for each c.c. of oxygen absorbed.

1 c.c. or 1 cub. centimetre = 0·061 cubic in.
∴ 3·3 calories per c.c.
= 93 large calories per cubic ft.
or 370 B.Th.U. per cubic ft. of oxygen absorbed.

This heat evolved when coal *absorbs oxygen* will be seen (on referring to Mr Graham's figure for the *adsorption* in coal of CH_4) to be between three and four times as great as the heat produced when *methane* is *adsorbed* by coal.

Mr Winmill,* however, considers this figure to be too great by about 50 per cent., and gives results of experiments which show the figure to be about 2·1 calories per c.c., *i.e.* 60 large calories or 237 B.Th.U. per cubic foot of oxygen absorbed.

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* *Trans. Inst. M.E.*, xlviii. 510.

MINING DEPARTMENT

CHAPTER IV.

MINE TEMPERATURES AND HUMIDITY.

There is no doubt whatever that the problem of temperature is quite as important from a mine-ventilation standpoint as that of the dilution of noxious and inflammable gases, and will prove more and more formidable as time goes on and deeper seams are sought. We may better understand the importance of the subject by treating it from a basis of causes and remedies.

The causes of heating of mine air may be briefly classified as follows :—

(*a*) The atmospheric temperature at the surface.

(*b*) The natural heat of the strata.

(*c*) The rise of air temperature as a result of auto-compression as it descends the downcast shaft.

(*d*) The heat given off by men, animals, lamps, machinery, etc.

(*e*) Underground fires.

(*f*) Oxidation of coal, timber, pyrites, etc., also their adsorption of air, oxygen, and other gases.

(*g*) Heat from crushing due to subsidence.

(*h*) The friction of the air against the surfaces of the roads, faces, etc.

(*i*) The heat disengaged by working operations.

Of these we may, for our purpose, omit any further reference to the last two causes.

(*a*) **The Atmospheric Temperature at Bank.**—The surface temperature, t, is, of course, the starting influence on the mine temperature, and if W be the algebraic sum of the heats disengaged or evolved, and absorbed by the air current up to any point, then the temperature, t_1, at that point can be expressed by

$$t_1 = t + \frac{W d_1}{c d_2} \quad . \quad . \quad . \quad . \quad (73)$$

where c is the specific heat of air and the ratio of the densities

above and below ground is $\frac{d_1}{d_2}$. This shows the influence of the bank temperature on the mine temperature, which in some cases calls for special temperature reduction apparatus, as we shall see later.

(*b*) **The Natural Heat of the Strata.**—It is known that down to a certain depth from the surface there is a zone the temperature of which is constant throughout the year. This zone, known as the " zone of equal temperature," is, in Great Britain, about 50 ft. deep, increasing about 1 ft. per degree increase in *latitude*.

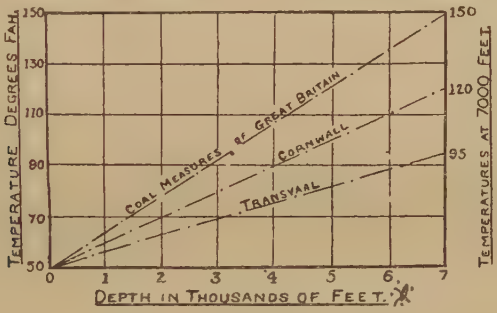

FIG. 40.—Geothermal gradient.

It is independent of climatic variations, and, like all strata zones bearing constant and uniform heat, is correctly described as a Geo-Isotherm. Below this superficial zone the strata increases in temperature. The rate of depth per 1° F. rise in temperature varies in different localities and often in the same geological formation in the same district, as will be seen by Table X. Thus for Great Britain it is about 64 ft. in the Midlands, 60 to 90 ft.* in certain parts of Yorkshire and the Scottish coalfield, 80 ft. in Lancashire and Kent, and about 100 ft. in Cornwall for 1° F. rise in temperature. In other countries it also varies greatly, being about 30 ft. in the Comstock Lode of Nevada, 140 ft. in parts of Brazil, and only 1° F. rise for 254 ft. of depth in the Village Deep Mine on the Rand.

On plotting depths against temperature, as in fig. 40, we obtain lines of varying steepness. This slope is known as the **Geothermal Gradient**. It has been found that highly inclined or broken strata have a less steep gradient than flat and even strata, which suggests that heat is more easily conducted along than across, or through,

* Penman, *The Mining Magazine*, June 1920.

the bedding planes. Strata having a low thermal conductivity will show a high geothermal gradient, which is one of the chief causes of the high gradient in some metalliferous mines.

It also depends on the surface contour, the lines of equal temperature being further apart under mountains than under valleys, which is due to the former being tensional and the latter compressional areas (see fig. 41). Another point in connection with the geothermal gradient is that there appears to be a critical depth for each set of conditions at which the uniformity of depth and temperature rise changes. At about 2000–3000 ft. the distance in depth often increases per degree rise. The method of obtaining the geothermal gradient, and also the associated temperature features, will be seen by carefully perusing Table X.

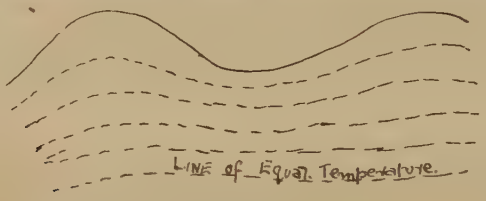

Fig. 41.

It will be seen that the effect of heating due to the natural heat of the strata is really a matter for deep mines, and will make big demands on the cooling effect of the ventilation. "In the downcast shaft and main intake airways of a well-ventilated coal mine the natural heat of the strata has no appreciable effect on the temperature of the air." *

The rise in temperature is chiefly at the working face, where the coal is freshly-exposed, and in order that the small temperature difference between the air in the downcast or main intake airways and the strata itself may be more fully recognised we will consider the case in the following manner :—

Weight for weight air and rock have very different heat-absorbing capacities. The specific heat of air is about 0·24, and of rock about 0·2, whilst a cubic foot of air weighs 36 grams and rock about 70,000 grams (154 lbs. approx.). The amount of heat required to raise the temperature of 1 cub. ft. of air 1° C. is $36 \times 0{\cdot}24 = 8{\cdot}6$ calories, and that for rock is about $70{,}000 \times 0{\cdot}2 = 14{,}000{\cdot}0$ calories, or in the ratio of about 1600 for rock to 1 for

* *First Report* on "The Control of Atmospheric Conditions in Deep and Hot Mines."

[To face page 94.

TABLE X.—*Summary of Most Reliable Strata Temperatures obtained in the Coal Measures.*

Position of test.	Date.	Position of thermometer in borehole.	Strata temperature.	Air temperature by borehole.	Height above sea-level.	Amount of cover.	Average surface temperature.	Geothermic gradient.	Remarks.
			Deg. Fah.	Deg. Fah.	Ft.	Ft.	Deg. Fah.	1° Fah. in ft.	
I.—YORKSHIRE.									
Brodsworth Colliery :									
Barnsley Seam . .	Oct. 1920	12 ft. in coal.	77·0	78·6	..	1824	48·0	62·9	Moderate gas-flow from borehole.
Duncil Seam borehole, No. 2.	July „	19½ ft. do.	79·0	75·4	122	1824	48·0	58·9	In recently exposed heading.
Parkgate Sinking No. 1	June „	8½ ft. in grey bind.	86·3	75·8	122	2160	48·0	56·4	Freshly exposed sinking.
Do. do. No. 2	Aug. „	8½ ft. do.	88·6	77·5	122	2272	48·0	56·0	Do. do.
Do. do. No. 3	Nov. „	8 ft. 7 in. do.	90·8	70·1	122	2554	48·0	59·7	Below Penton Seam.
Penton Seam . .	Oct. 1921	15½ ft. in coal.	91·6	76·6	122	2540	48·0	58·5	50 yards from shaft.
Bentley Colliery :									
Barnsley Seam . .	Feb. „	18 ft. do.	74·5	70·0	16	1809	48·3	69·0 [1]	In coal-face.
Do. do. . .	April 1919	8 ft. 4 in. in coal	78·8	76·0	16	2046	48·3	67·0	Do.
Neighbourhood of Barnsley Seam.	Jan. 1920	66 ft. in grey bind.	61·6	73·0	16	2167	48·3	64·7	Vertical borehole.
Above Barnsley Seam .	Feb. 1921	35½ ft. do.	75·5	65·0	16	1800	48·3	66·0	Borehole put in side of main intake; exposed to air current for about 8 years.
Yorkshire Main Colliery :									
West side in Barnsley Seam.	Mar. „	19½ ft. in coal.	93·6	85·5	..	2742	48·1	60·3	This portion of heading exposed for 10 days to a ventilation of 1200 cub. ft. of air per min.
East side in Barnsley Seam.	Nov. „	21 ft. do.	90·1	85·0	70	2500	48·1	59·5	Exposed for 7 days before boring to a current of 2260 cub. ft. of air per min.
South above Barnsley Seam.	Mar. „	32½ ft. in grey bind.	88·7	..	..	2615	48·1	64·0	In main intake road; exposed for 9 years; gradient may be low.
Hatfield Colliery :									
(A) No. 1 Stone Drift .	Aug. „	32½ ft. do.	89·5	87·1	13	2425	48·3	58·8	About 42 yards above Barnsley Bed; ventilation, 1600 cub. ft. of air per min.
(B) Barnsley Seam .	„ „	25 ft. in coal.	89·9	89·4	13	2425	48·3	58·3	Moderate gas-flow; ventilation, 12,130 cub. ft. of air per min.
(C) Fault-face . .	„ „	20½ ft. in grey bind.	85·8	86·6	13	2298	48·3	61·3	Hole in fault face; level of Kent Thick Coal.
(D) Kent Thick Seam .	„ „	20 ft. 4 in. in coal.	85·5	82·0	13	2302	48·3	61·9	Very little gas given off; ventilation, 688 cub. ft. of air per min.
Throne Colliery :									
Vertical borehole, 16 E .	April „	1200 ft. vertical in water	61·5	..	7	1200	48·3	91·0	By No. 2 shaft; 4½-in. hole full of water.
	„ „	1350 ft. do. do.	62·8	..	7	1350	48·3	93·0	
	„ „	1500 ft. do. do.	64·8	..	7	1500	48·3	91·0	
Do. do. 15 W.	„ „	1200 ft. do. do.	61·5	..	7	1200	48·3	91·0	By No. 1 shaft; 4½-in. hole full of water.
	„ „	1350 ft. do. do.	63·0	..	7	1350	48·3	92·0	
	„ „	1500 ft. do. do.	64·8	..	7	1500	48·3	91·0	
II.—FIFESHIRE.									
Bowhill Colliery :									
Five Feet Seam . .	July–Aug.	18 ft. in coal.	66·8	73·4	278	1388	47·0	70·1	
Do. do. . .	„ 1919	17 ft. do.	73·7	79·4	306	1764	47·0	66·1	High-temp. reading due to neighbourhood of old workings. Probably not true gradient.
Do. do. . .	„ „	17 ft. do.	79·1	77·8	333	2387	47·0	74·3	
Do. do. . .	„ „	6½ ft. do.	81·9	81·6	325	2814	47·0	80·7	
Do. do. . .	„ „	17 ft. do.	82·0	80·6	325	2815	47·0	80·5	
III.—LANCASHIRE.									
Pendlebury Colliery :									
Rams Seam . .	Nov 1919	30½ ft. in coal.	65·1	64·0	240	962	49·0	60·0	
Do. do. . .	„ „	25 ft. do.	65·2	62·2	240	982	49·0	59·5	
Four Foot Seam . .	„ „	19½ ft. do.	77·3	81·8	240	1936	49·0	68·4	Gas-flow inappreciable.
Rams Seam . .	„ „	15½ ft. do.	83·1	87·1	240	2305	49·0	67·6	Do. do.
Dow Seam . . .	„ „	20½ ft. do.	87·0	82·0	240	2322	49·0	61·1	Do. do.
Dow Seam . . .	Dec. „	8 ft. 9 in. do.	88·1	84·88	240	2391	49·0	61·1	Do. do.
Pendleton Colliery :									
West Side, Rams Seam	„ „	18 ft. do.	100·0	100·0	124	2450	49·4	69·0	Ventilation, 1600 cub. ft. of air per min.
East Side . .	„ „	14 ft. do.	98·3	98·0	124	3503	49·4	71·6	Ventilation, 1300 cub. ft. of air per min.; gas-flow inappreciable.
IV.—NORTH STAFFORDSHIRE.									
Racecourse Pit :									
Great Row Seam .	Dec. 1920	29½ ft. in coal.	58·4	56·2	..	930	47·5	85·0	Coal exposed for 6 months; ventilation, 1800 cub. ft. of air per min.; gas-flow inappreciable; cooling effect of this negligible at 29 ft.
Deep Pit :									
Mossfield Seam . .	Feb. „	21½ ft. do.	77·1	73·3	486	1956	47·5	66·1	No appreciable gas-flow.
East above Ragman Seam.	„ „	22 ft. in grey bind.	80·6	82·0	..	2103	47·5	63·5	In side of return airway.
Holly Lane Seam .	„ „	23½ ft. in coal.	97·6	91·0	547	3168	47·5	63·2	No gas-flow from borehole.
Cross-measure Dip .	„ „	19 ft. 2 in. in grey bind.	102·3	88·0	470	3323	47·5	60·6	In side of road.
Cockshead Seam . .	Mar. „	28 ft. 10 in. in coal.	105·5	87·0	513	3475	47·5	59·9	
V.—SOUTH STAFFORDSHIRE.									
Hamstead Colliery :									
Ten Yard Seam . .	Mar. 1922	19½ ft. in coal.	68·3	72·0	400	2120	48·3	106·0	Heading driven past this point some days previously; no appreciable gas-flow.
Holly Bank Colliery :									
Eight Feet Seam .	Apr. 1921	20 ft. do.	73·3	80·81	520	2391	48·0	94·5	Gas-flow only moderate.
VI.—CUMBERLAND.									
Ranney Gill borehole, Brayton.	May 1922	1560 ft. in vertical borehole.	69·3	47·3	..	1560	47·3	70·9	Borehole filled with water; error from convection unlikely; internal diameter of tube, 3⅞ in.
		2160 ft. do. do.	78·7	47·3	..	2160	47·3	68·8	Internal diameter of tube, 3¼ in.
		2726 ft. do. do.	90·0	47·3	..	2726	47·3	63·8	Internal diameter of tube, 2 in.

[1] This figure for gradient may be low, owing to general cooling of coal by evolution of gas.

From a paper by J. Ivon Graham, M.A., M.Sc., for the Committee Investigating the "Control of Atmospheric Conditions in Deep and Hot Mines."

air on a volume basis. Assuming that the air, in traversing 1000 ft. of roadway 50 sq. ft. in cross-sectional area in one minute, sustains an increase in temperature of 2° C. (3·6° F.) by the absorption of heat from the strata, it will derive $\frac{1}{5·6} \cdot 36 \cdot 0·24 \cdot 2 = 0·345$ calories per foot traversed. This abstraction will cool 1 cub. ft. of the rock by $\frac{0·345}{1600} = 0·00021°$ C. If the abstraction has penetrated to any distance it will affect a corresponding thicker stratum, so that if we take the penetration to be only 1 ft. deep, *i.e.* about 30 cub. ft. per foot run for the above airway, the rock will be cooled at the rate of only $\frac{0·00021}{30} = 0·000007°$ C. So for short distances and periods the temperature difference between fresh air and rock is practically constant. Duration of contact, *i.e.* velocity of the air, is the deciding factor governing the degree to which the air is heated, which velocity, since it is high in the main intakes and shaft, is one great cause of the small influence of rock temperatures on the temperature of the air in these places. The *change* in temperature sustained by the air current in a given length of airway is decided by the density of the air and the sectional area of the airway, and is greater in proportion as the density is lower. The greater the area of contact between air and rock the larger the amount of heat transmitted from one to the other: it is therefore advisable that the intake airways should be of circular or square cross-section, as affording the most favourable ratio between superficial area and cubical capacity. Excessive splitting or dividing of the air current favours heating by lowering the velocity and rate of heat abstraction. Frequent reversal of the current also tends to increase the temperature of the air.

(*c*) **The Rise in Temperature Due to Compression in the Downcast.**—The adiabatic compression of the air itself in the downcast will set up heat, and this amount has been found to be * 5½° F. rise for each 1000 ft. of depth, or 1° F. for each 182 ft.

Professor Galloway deduces this figure by considering the case of the moving vertical column due to ordinary exhaustive ventilation in shafts, and disregarding for the purpose of discussion any gain or loss of heat from other sources. Assuming a uniform temperature of the atmosphere, to which the column is open, its volume will decrease and its temperature increase (in accord-

* Dr Galloway, *Lectures on Mining*, and *Report* (Cd. 2091) to Home Secretary by Messrs Haldane, Martin, and Thomas.

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(*c*) **The Rise in Temperature Due to Compression in the Downcast.**—The adiabatic compression of the air itself in the downcast will set up heat, and this amount has been found to be * $5\frac{1}{2}°$ F. rise for each 1000 ft. of depth, or 1° F. for each 182 ft.

Professor Galloway deduces this figure by considering the case of the moving vertical column due to ordinary exhaustive ventilation in shafts, and disregarding for the purpose of discussion any gain or loss of heat from other sources. Assuming a uniform temperature of the atmosphere, to which the column is open, its volume will decrease and its temperature increase (in accord-

* Dr Galloway, *Lectures on Mining*, and *Report* (Cd. 2091) to Home Secretary by Messrs Haldane, Martin, and Thomas.

ance with the laws of Boyle and Charles) in proportion to the depth below the surface. The movement is due to the manometrical depression created by the ventilator at the base of the column, and this movement ceases when the ventilating force stops. Thus the change of temperature is proportional to the work done, which latter is, therefore, an exact measure of the temperature change. Consequently a mass of air of sp. ht. s, weight d, and initial absolute temperature T_1, will acquire an absolute temperature T_2 as a result of descending a shaft D ft. deep. The work done in compressing the air will, therefore, be equal to the work done in lowering the air to this depth.

$$\therefore 778ds(T_2 - T_1) = dD \therefore T_2 - T_1 = t = \frac{D}{778s} \quad . \quad (74)$$

E.g. assuming that the sp. ht. of air at constant pressure is 0·238, what is the temperature rise due to autocompression in a shaft 2000 ft. deep ?

By (74) above $$t = \frac{2000}{778 \times 0{\cdot}238} = 10{\cdot}82^\circ \text{ F.}$$

From the foregoing it will be seen that the thermal effect of compression in the shaft is 5·4° F. per 1000 ft., say $5\frac{1}{2}$, *i.e.* 1° F. rise for every 182 ft. depth.

The student may here have difficulty in understanding that any compression occurs in the downcast, knowing that the fan situated at the top of the upcast is exhausting air from the downcast through the workings and upcast shaft up to the fan drift. It must be noted that whatever barometric pressure is obtained under these conditions it is the result of the total compression in the shaft minus the effect produced by the ventilator. The biggest ventilating pressures found in mines are equal to about 12 in. of water column, whilst the normal atmospheric pressure is about 34 ft. of water column. Therefore the effect of the fan on the atmospheric pressure as read by the barometer is very small, and would not be very appreciable. The fan, if stopped, would in the above case only increase the barometric column in the ratio of about 1 : 34 approx. Now the water-gauge, W.G., is a differential gauge, and not an absolute gauge like the barometer, so the effect of the above stoppage on the water-gauge would be pronounced, which effect is aided by the fact that the density of the medium is 13·6 times less. The compression in the downcast is compensated by an equivalent

expansion in the upcast, but here the cooling effect of expansion is wasted. The effect on a shaft 2500 ft. deep is to increase the mine temperature at the bottom of the downcast by $\frac{2500}{182}$, or about 13° F. before entering the workings. Of course expansion will occur as the air ascends the rise workings which will give some cooling effect, but the major portion of this effect is wasted in the upcast.

(*d*) **The Heat given off by Men, Animals, etc.**—Although the human body-temperature is extremely steady at about 98° to 99° F., any abnormality in it causes symptoms of illness. It is well known that men can live in perfect health at air temperatures varying from well below zero up to 120° F. or more, and probably no animal can stand such variations in external temperature and retain its physiological efficiency so well as man. The state of the humidity of the air affects the amount of heat given off by men and animals. The amount of heat given off by a man in radiation and conduction, whilst working, is under fair conditions about 250 calories per hour throughout the shift. If at rest the amount is only 70 calories, so we may state that the effect of this on the general ventilation is small; but in confined places the effect would soon become appreciable. Lights, like man, are less efficient when forced to retain their own heat and are also a source of heat due to their own rise in temperature, even up to several hundred degrees F.

(*e*) **Underground Fires.**—The influence of mine fires, especially gob fires, is often so serious as to necessitate the sealing off of valuable and extensive workings. Their effect on the mine air temperature will depend on their position in the air circuit, extent, etc. The temperature in these cases will be conveyed by conduction through the strata, and in bad cases radiation will be considerable, as also convection. Careful checking with suitable instruments, as in figs. 42 to 44,* must always be resorted to, and also many precautions, such as eliminating combustible matter from wastes, and other measures too numerous to be detailed here. Since gob fires are more or less sporadic in character and also detachable from the main air currents, their influence on mine temperatures will not be constant, and will need no further mention under general causes of heating in mines.

(*f*) **Oxidation.**—This is the greatest cause of heating in mines, and were heat from this source not absorbed by other agencies

* See Newton, "Goaf Heatings," *Trans. Inst. M.E.*, 1922.

Fig. 42.

the air temperature would soon rise to above the boiling-point of water, long before which conditions would have become untenable. To elucidate this we will assume that one-hundredth of a cubic foot of oxygen is being absorbed by the coal for each foot run of coal face, and that 10,000 cub. ft. of air passes through the face per minute.

Fig. 43.

Accepting the lower figure of the previous chapter as given by Mr Winmill for the thermal effect of oxidation, *i.e.* 237 B.Th.U. per cubic foot of oxygen absorbed, then with air weighing 0·0766 lbs. per cubic foot (sp. ht. = 0·24 ft.), we obtain a rise in temperature of $\frac{237}{10000 \times 0{\cdot}0766 \times 0{\cdot}24} = 0{\cdot}0129°$ F. for every foot of face traversed by the air. This would certainly become very great in a very short space of time were it not for the many cooling agencies which will be mentioned later.

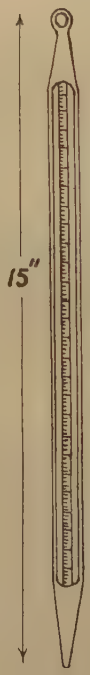

Fig. 44.

Dr Haldane * found that certain coals evolve heat, due to oxidation, sufficient to raise the temperature of mine air 2·65° F. for each diminution of 0·01 per cent. of oxygen from the air current. From this we can look at the above discussion from another view-point, arriving ultimately at a result somewhat similar. To cause a loss of 1 per cent. of oxygen will mean a rise of air temperature of 265° F., to which the same remarks apply as to the above. We have also seen in the previous chapter that, apart from oxidation by coal, timber, and strata, coal is capable of adsorbing methane, which

* *Investigation of Mine Air.*

sets up considerable heat, amounting to about 110 B.Th.U. per cubic foot adsorbed. Though at coal faces the reverse effect, *i.e.* liberation of gas, is commonest, we might expect very considerable heating from this cause in addition to the above in wastes, old workings, and other places where finely disseminated or broken coal might be expected to exist.

Calculation of the thermal effect of oxidation becomes very complex at higher temperatures, the oxidation becoming self-propellent at 200°–275° C.*

(*g*) **Crushing Due to Subsidence.**—When strata are crushed in the process of subsidence mechanical work is converted into heat, and this heat is distinct from any subsequent heat which may be developed as a result of the oxidation of the newly-exposed particles. There are two main arguments as to the law of heating by crushing or grinding. These theories are known by the names of their originators, the eminent mining physicists, Kick and Rittinger; both are the subject of much controversy.† Kick maintained that the energy expended in crushing is proportional to the amount of subdivision the individual fragments undergo; Rittinger, the earlier theorist, says that the energy consumed in fracture is proportional to the area of new surface created. This view of Rittinger has been contested by S. J. Speak, President, Inst.M.M.,‡ on the grounds that the tensile force required to fracture a member is greater as the member is longer, and proceeds to show that the same quantity of work is required to break a wire would suffice to break again both portions of the wire (neglecting the elongation and the reduction of the area at fracture). In other words, we may get two fractures of the shorter wire for practically the same amount of work as was expended to make one fracture of the original wire, thus defining in a practical manner the meaning of "work," *i.e.* force acting through a distance. Continuing this reasoning indefinitely, we see that the work needed to produce one particle of 1 cub. in. is the same as that needed for n particles each of 1 nth cub. in. Kick enunciates this as follows: "The energy required for producing similar changes in shape in similarly shaped bodies of equal state varies as the volumes or weights of these bodies." Practice does not appear to bear this out, probably as a result

* *Illinois Univ. Bull.*, No. 16, and Prof. Bone.

† *Engineering and Mining Journal Press*, January 1923.

‡ S. J. Speak, "The Energy Consumed in Crushing," *Mining Mag.*, January 1923.

of the low theoretical efficiencies of crushing machines. On the other hand, Professor Briggs * cites many cases supporting the surface theory of Rittinger. It appears to be probable that conditions supporting both these theories may concur, especially when one considers the great differences in the extent and character of cleavage of the various rocks, to cite only one aspect of the problem. Most minerals and rocks (having a definite atomic arrangement deciding the cleavage planes irrespective of the size of fragment) may be expected to be more consistent respecting a law of crushing than coal which has variable cleats, but at the same time soon produces great surface area in the small state. Viewing this and other factors, Professor Briggs attacks the problem along Rittinger's lines from the standpoint of the relation of the work done to the resulting surface and size reduction, the figures for which must of necessity be rough approximations. Consequently he finds from a first principles consideration that

TABLE XI.—*The Thermal Characteristics of Strata Encountered in Coal Mining.*†

Name.	Specific heat. *s.*	Heat diffusivity (C.G.S. units). K.	Heat conductivity (C.G.S. units). *k.*
Bituminous coal	0·2–0·4	0·002	0·00068
Cannel coal	0·26	0·004	0·0012
Coal measures shale	0·2	0·005	0·00235
Slate	0·21	0·0073	0·00327
Carboniferous limestone	0·217	0·0095	0·0055
Magnesian limestone	0·245	0·0082	0·00522
Sandstone	0·229	0·01	0·00672
Basalt	0·202	0·0093	0·0052
Granite	0·195	0·011	0·00584

if u be the work done per square inch of surface created during crushing a cube of rock of A in. size and of specific gravity d down to a size of a in., then the work done in crushing one pound of material will be—

$$\frac{167u}{d}\left(\frac{1}{a}-\frac{1}{\mathrm{A}}\right)\text{ft.-lbs.} \quad . \quad . \quad . \quad (75)$$

* Henry Briggs, "Possibility of Spontaneous Combustion, etc.," *Trans. Inst. M.E.*, vol. lxiv.

† *Ibid.*

and if the ratio of reduction in size is great—

$$\frac{167u}{da} \quad . \quad . \quad . \quad . \quad . \quad (76)$$

and therefore the heat developed per pound of mineral

$$= \frac{167u}{777da} \text{ B.Th.U.} \quad . \quad . \quad . \quad (77)$$

and per square inch of surface created $= \frac{u}{777}$ B.Th.U.

This is found to be 0·00036 B.Th.U. per square inch of fractured surface for bituminous coal. Knowing that crushing heat is primarily a surface effect, and calling the above quantity of heat q, a particle of surface S, weight W (w per inch), and specific heat s, will receive an amount of heat Sq, and the temperature rise T sustained will be—

$$\text{Temperature rise} = \frac{\text{heat produced}}{\text{weight} \times \text{sp. ht.}} = \frac{Sq}{Ws} = \left(\frac{Sq}{ws}\right)\frac{1}{a} = T \quad . \quad (78)$$

and if the fragment be spherical it can be shown that the temperature throughout it will be—

$$\frac{Sq}{dsa} \ (a \text{ being the radius}) \quad . \quad . \quad . \quad (79)$$

The heat generated in fragmental bodies is a surface feature and attains the interior by the proximity of other particles. It is quicker and more intense in harder rocks, so that the above expression of the temperature throughout a crushed fragment is deduced from a discussion of the heat diffusivity and conductivity of the rocks * in question and the fineness of the particles. Really this is combining the theories of Kick and Rittinger.

The valuable conclusions of Professor Briggs respecting the heat developed due to crushing are, in effect, as follows :— †

1. The heat produced by crushing is one of the causes leading to spontaneous combustion in mines.

2. In most cases the direct effect is to pave the way for accelerative oxidation.

3. The smaller the fragment the greater the temperature rise and the greater the surface exposed to oxidation.

* See "Definitions," Chapter I.

† Henry Briggs, "Possibility of Spontaneous Combustion, etc.," *Trans. Inst. M.E.*, vol. lxiv.

4. A fine fragment if suddenly crushed among others from $\frac{1}{300}$ in. will obtain a temperature rise of 52° F., and from $\frac{1}{100}$ in. 17° F.

5. The heat being generated at the fractured surfaces, the particle, if small, has practically the same temperature throughout.

6. Considerable temperatures are produced by crushing hard rocks suddenly and finely. Coal measures shale has a 50 per cent. greater temperature rise than bituminous coal.

Remedies or Cooling Influences which tend to reduce the temperatures in mines may be briefly summarised as follows :—

(*a*) Vaporisation of the water formed by oxidation.

(*b*) The expansion of the air on becoming heated.

(*c*) The emission of gases from the strata.

(*d*) The evaporation of moisture in the air by means of suitable velocities.

(*e*) Practical mining methods.

(*f*) Mechanical methods.

(*a*) **Vaporisation of the Water formed by Oxidation.**—The latent heat of vaporisation of water, *i.e.* the amount of heat required to turn it all into vapour, is tremendous. One kilogram (2·2 lbs.) of water requires 582 calories, or roughly about 1000 B.Th.U. per pound to convert it into vapour under the conditions existing in the average mine. Assuming a mean of the proportions of water formed by the oxidation of coal to contain 35 per cent.* of the absorbed oxygen as constituting the water, we may compute the thermal effect of vaporising this amount. In each cubic foot of oxygen absorbed 0·35 cub. ft., *i.e.* 0·028 lb., goes to form water.

$$0{\cdot}028 \text{ lb. oxygen} \cdot \left(0{\cdot}028 + \frac{0{\cdot}028}{8}\right) - 0{\cdot}0315 \text{ lb. of water which}$$

needs for vaporisation $1000 \cdot 0{\cdot}0315 - 31{\cdot}5$ B.Th.U. This heat is absorbed by the vaporisation which will reduce the thermal effect of oxidation to approximately 200 B.Th.U. This effect is small and depends on the factors governing vaporisation, *i.e.* humidity, velocity, temperature, etc.

(*b*) **The Expansion of the Air as it becomes heated.** -The counter-effect of the compression of the air in the downcast is obtained by the expansion in the upcast, which here is of small benefit as a cooling influence on the workers in the mine. To utilise this expanding effect to the best advantage the fresh air

* Graham, *Trans. Inst. M.E.*, lii. 353.

should be taken direct to the lowest parts of the workings. From here the expansion will assert itself and set up a slight cooling effect, as its density diminishes with increase of temperature and altitude, so causing it to tend to move toward the rise workings, thus also assisting the ventilating current.

(*c*) **The Emission of Gases from the Working Face and Strata.**—It was seen in the preceding chapter that heat is evolved when oxygen and methane are absorbed and adsorbed by coal: we should expect a reverse effect of equal magnitude when these gases are liberated from the coal. This actually occurs, the heat being re-absorbed from its surroundings, when the gases are liberated, as at the working faces. It is important to note here that were the gases simply compressed into the coal the cooling effect on liberation would be trifling: the gases being liquefied on the internal surfaces of the coal in the process of adsorption have, therefore, a latent heat of evaporation which produces a considerable cooling effect. Very gassy longwall faces for this reason are often appreciably lower in temperature than the strata in their vicinity. By the above we do not mean that the faces are the coolest parts of the mine, in fact they are the places where the greatest temperature rise occurs (see figs. 45 and 46). What is meant is that the cooling effect is very appreciable at the face, and here it has greatest need to be so. It balances in some measure the oxidation heating effect.

(*d*) **The Evaporation of the Moisture in the Air.**—This is the greatest cooling influence in the mine. The amount of moisture in the air increases with the temperature, and, as stated in former chapters, the saturation point can be increased by increasing the temperature. Column 3, Table VI., will convey to anyone an idea of this moisture content of the atmosphere. The air if unsaturated is always tending towards saturation, and since in mines the air temperature increases as the current proceeds through the workings, its capacity for absorbing more moisture is also being increased. The amount of moisture removed from a mine in a day by means of the ventilator alone is almost incredible, as the following example will show:—

E.g. 200,000 cub. ft. of air per minute leaves a mine at 90° F., and the relative humidity is 56 per cent., find the weight of water removed by the air daily.

Col. 3, Table VI.—

1 cub. ft. of air at 90° will hold 0·00213 lb. of moisture when saturated.

1 cub. ft. of air at 90° will hold $\frac{0{\cdot}00213 \times 56}{100}$ lbs. of moisture 56 per cent. saturated.

$\therefore$ 200000 cub. ft. of air at 90° will hold $\frac{200000 \times 0{\cdot}00213 \times 56}{100}$ lbs. of moisture when 56 per cent. saturated

$= \frac{200000 \times 0\ 00213 \times 56 \times 24 \times 60}{100 \times 2240} = 153{\cdot}36$ tons daily or over 6 tons per hour.

This humidity prevents evaporation, and therefore cooling, and affects the working capacity of men. This effect then depends not on the dry-bulb temperature of the hygrometer but upon the wet-bulb temperature. This will be seen when one notes that radiation and conduction away of the heat caused by physical exertion is the means whereby a person's body is kept cool. These means may be either (1) deliberate means, like the adoption or the discarding of clothing according to whether one feels cold or hot; or (2) involuntary means, chiefly by the perspiration of the body. The perspiration evaporates and so a cooling effect is set up. If the air is already saturated it can take up no more vapour from perspiration or any other source. The evaporation cannot then take place, and so the body temperature rises to the detriment of the working efficiency of the miner. Thus we see that it would not matter how much a man perspired he would not become cooled in a saturated atmosphere because the perspiration would not evaporate but simply drop from him as waste. The resultant high temperature does not appear to injure the health; it only affects the working capacity of a man and increases his discomfort. If the saturation were even at a lower temperature the body could still keep cool if only the rate of evaporation was fast enough. This rate of evaporation is proportional to the rate at which the air passes over the body, *i.e.* to the velocity of the air current. Now the wet-bulb temperature is that to which a moist surface can be cooled by evaporation in air, therefore the effect of warm air in raising the body temperature abnormally will depend on the wet-bulb as stated above. Dr Haldane * has found that in still air, without working, 88° F. is the maximum wet-bulb temperature at which a man feels comfortable, even with a minimum of clothing, though short intermittent work, supervisory tasks, and other

* *Journal of Hygiene*, v, 494.

work not of an arduous nature has been done in temperatures as high as 94° F. wet-bulb. The dry-bulb does not matter much: an air temperature of 133° F. having a wet-bulb of 88° F. has no more effect on the body temperature than if both bulbs registered 88° F. We may safely state that about 80° F. wet-bulb is the

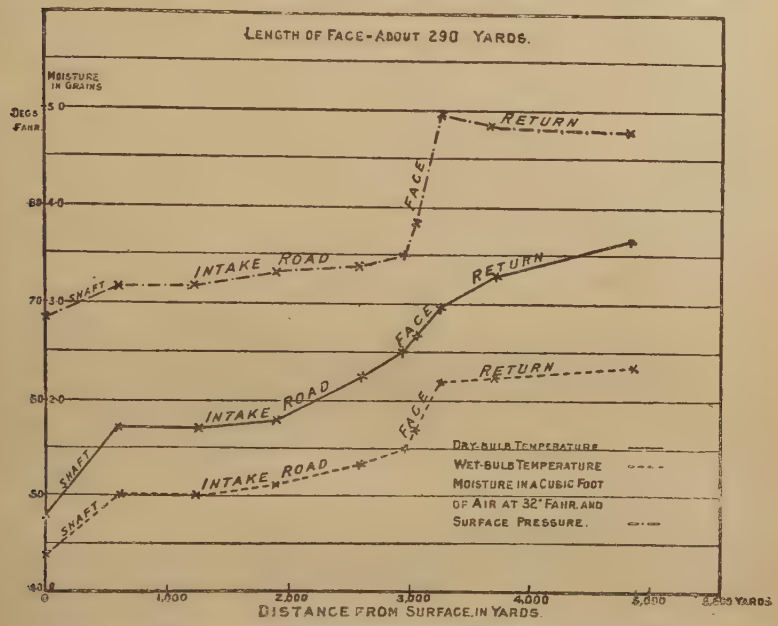

FIG. 45.—Chart of temperature and moisture content, Bentley Colliery, April 1919.

limit for hard manual work, but continuous hard work all day is impossible at 78° F. wet-bulb.

Among other adverse physiological effects due to humid atmospheres may be mentioned the disease known as ankylostomiasis, due to the breeding of the intestinal worm *ankylostoma duodenale* within the patient. The worm is extremely prolific, the ova being deposited in the intestines and hatched outside the body in human dejections, the most favourable conditions for which are warmth, moisture, and darkness. The goafs and old workings of humid mines obviously provide these conditions. The parasite destroys the functions of the blood and causes much bleeding, which develops anæmia and, if allowed to develop further, ultimately causes death. Fortunately the disease is rare

in England, and outside of Cornwall very few cases have been recorded, but in some of the humid mines of Belgium and the tropics it often becomes serious. Dry, cool atmospheres are the essential preventatives.

The chart * (fig. 45), taken at Bentley Colliery, near Doncaster, illustrates well the effects of good ventilation on the wet-bulb temperature and the taking up of moisture by the air. It will be seen that a rise of 9° F. occurs in the shaft due to the auto-compression of the air. The first 1300 yards of intake show only 1° F. rise in temperature. The increase of the temperature thereafter is due to splitting the current, and it continues to rise until 1000 yards from the upcast shaft. The rise is rapid along the face of the freshly-exposed coal, due to oxidation. This occurs in all mines, and would be much greater but for the cooling influences previously discussed. The wet-bulb, it will be seen, in no place tends to approach the dry-bulb temperature, but actually diverges from it. This lag of the wet-bulb means that the relative humidity is decreasing, and therefore the probability of inconvenience to the workmen due to lack of cooling effect is being lessened. Dry roads allowing air with small moisture content to be taken right up to the face are the cause of this, so that the wet-bulb is kept low and the cooling capacity correspondingly high.

Where the mains give off much heat and the roads are wet, and also where the current is sluggish owing to long coursing, small quantities, or excessive splitting, the reverse effect from the Bentley example above, is to be expected. Such an example is provided by the Bowhill Colliery, Fife, where the two bulbs are almost identical in reading, as shown in the chart,† fig. 46. The high relative humidity is due to the shaft being so small that the quantity is very restricted, being only 40,000 cub. ft. per minute. Also there is much splitting of the air, leaving only about 6000 cub. ft. for the last 1000 yards of intake. The intake is very hot, and so the air collects much moisture, a fog being present in the mine in many places.

(*e*) **Practical Mining Methods of Cooling.**—Among the practical methods in the mine for dealing with the current from a cooling standpoint, driving the intakes in the solid rock strata has been

* *First Report* of the Committee on "The Control of Atmospheric Conditions in Deep and Hot Mines."

† *Third Report* of the Committee on "The Control of Atmospheric Conditions in Hot and Deep Mines."

suggested. This idea is sound, since rock absorbs less oxygen than coal or shale, therefore the current would be cooler. This method has the disadvantage of being very expensive. The same remarks apply to the suggestion for lining existing roads in coal or shale with brickwork, concrete, ashes, or stone. Increasing the volume of air passing by making splits at the proper places and in similar proportions and the adoption of regulators where a small saving in air can be effected are also advantages.

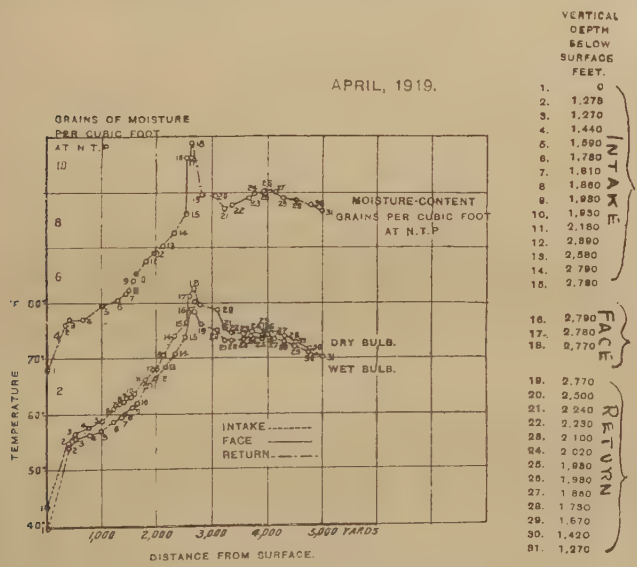

FIG. 46.—Bowhill Colliery, Fife.

In this way a greater volume, and therefore velocity, can be obtained in other districts. The prevention of leakage should aid this purpose greatly. Stone dusting will also have a beneficial effect on the amount of heat produced.

Liquid air has also been proposed as a method of cooling. A single gallon of liquid air is capable, by its evaporation from liquid to air at 70° F., of abstracting 1800 B.Th.U. of heat, *i.e.* enough to cool 7000 cub. ft. (approx.) of air to 10° F. In so doing it expands one thousandfold. Apart from its cooling effect it enriches the oxygen content in bad atmospheres.

In certain dry mines, like the City Deep Rand Mine, spraying with water may even be adopted.

(*f*) **Mechanical Methods of Cooling Mine Air.**—Climatic conditions greatly affect the system of ventilation in mines, especially in very dry or very humid areas, as in foreign and colonial coalfields. Mr E. H. Clifford has shown that, at the City Deep Mine on the Rand, the temperature of large volumes of dry air are not heated to the same extent there as in England owing to the very low geothermal gradient (1° F. = 254 ft.). Thus water can be introduced into the workings, to keep down the dust from the rock drills (it being a gold mine), and also to cool the air by spraying, without a risky rise in the wet-bulb reading. On the other hand, Mr Eric Davies states that though the geothermal gradient is low (1° F. = 140 ft.) in the St John del Rey Mine, Brazil, and the mine is also dry, the climatic conditions are very favourable for high humidity and saturation. So much is this so that a refrigeration plant has been adopted in order to cool the intake air at the surface, thus causing it to release some of its moisture by condensation, so drying it.

This plant * being in its way unique and well illustrating some of the principles of this chapter, will be dealt with here in some detail. It was constructed with a guaranteed capacity that it would have a heat extraction rate of 100,600 B.Th.U. per minute, corresponding to a reduction of 72° to 43·2° F. wet-bulb (which is a saturated moisture content fall from 58 to 42 grains per cubic foot), and to act at any degree of saturation of 5040 lbs. of dry air per minute, *i.e.* about 80,000 cub. ft. per minute at the density prevailing at the mine. The plant is constructed on the cold-storage, brine-circulation system, but water is used instead of brine, and air coolers instead of a cold-storage chamber. The plant (fig. 47) is divided into six stages to meet the varying seasonal duties, and the progress of the successive stages is as follows:—

In stage	I.	the air is cooled	from	72°	wet-bulb	to	67·6°	Fahrenheit.
"	II.	"	"	67·6°	"	"	62·6°	"
"	III.	"	"	62·6°	"	"	57·6°	"
"	IV.	"	"	57·6°	"	"	52·6°	"
"	V.	"	"	52·6°	"	"	47·8°	"
"	VI.	"	"	47·8°	"	"	43·2°	"

Thus initial stages can be shut down when the wet-bulb temperature is below their figures, as given above; on cold winter

* "Air-Cooling Plant at the Morro Velho Mine of the St John del Rey Mining Co., Ltd., Brazil," by Mr Eric Davies, *Trans. Inst. M.E.*, June 1922; *Iron and Coal Trades Review*, 23rd June 1922.

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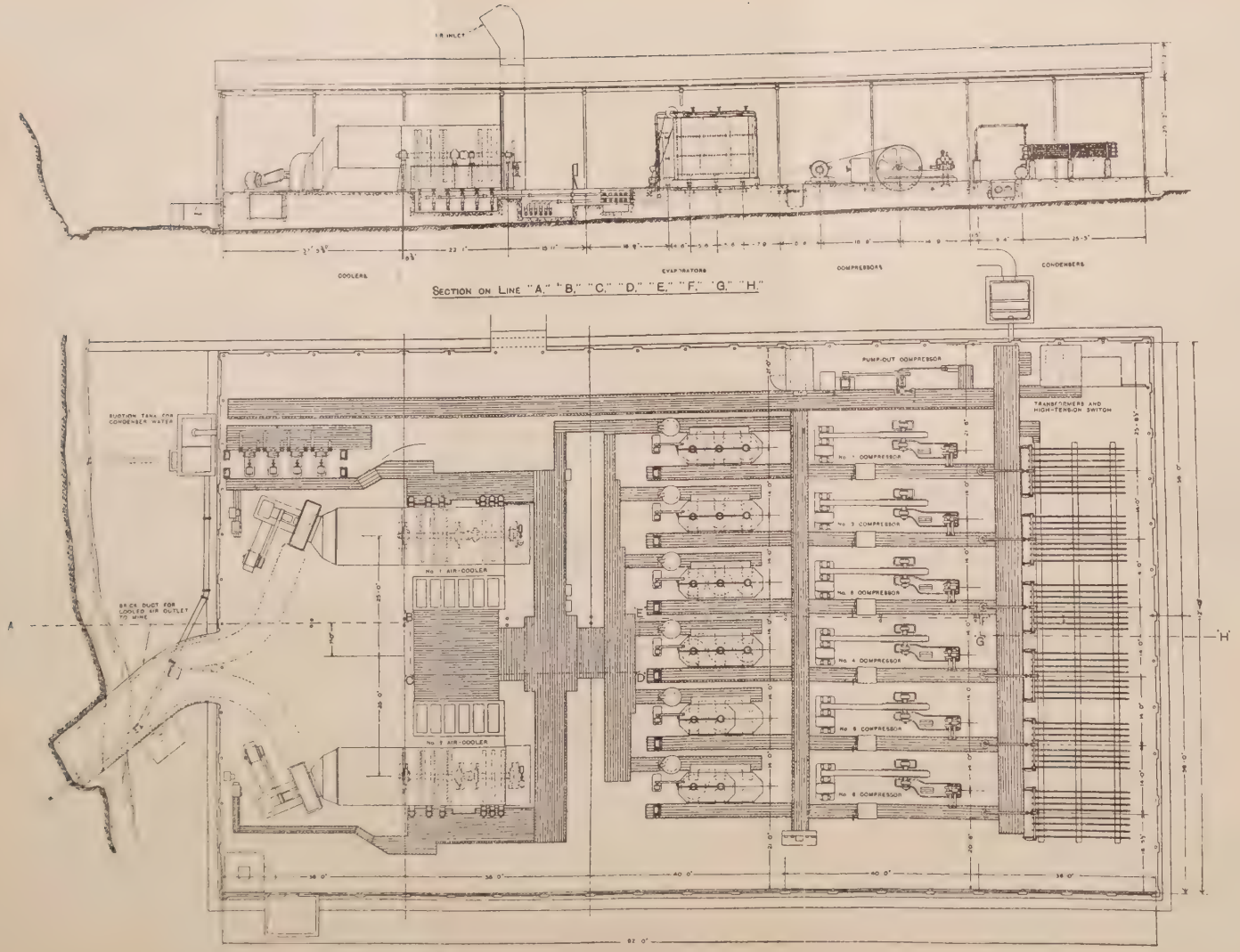


FIG. 47.—Air-cooling plant at Morro Velho Mine, Brazil.
(From paper by Mr Eric Davies, read before Inst. M.E., 1922.)

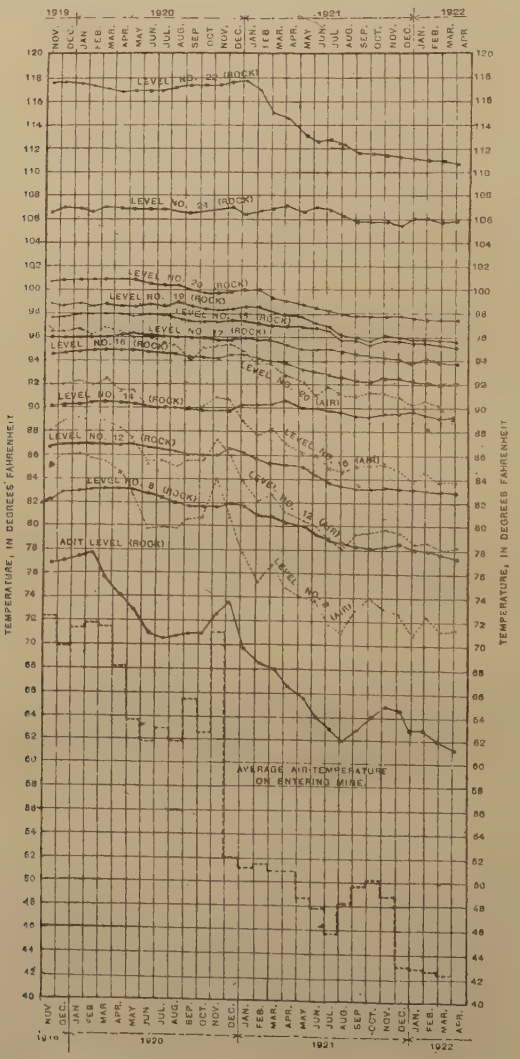
nights the whole plant can be shut down, for then the initial wet-bulb is below 43·2° F.

Fig. 47 on the opposite page shows the general layout of the plant. On the left of the figure will be seen the air coolers, of which there are two sets, acting in parallel, so that each deals with about half the total quantity of air. The air is pulled through the coolers by means of two 60-in. fans, belt-driven at 400 r.p.m. by a 50-h.p. variable speed motor, delivering into a common duct below the floor level for the downcast. These coolers, made by Heenan & Froude of Worcester, consist of a number of 10 ft. 10 in. diameter cylinders, 8 ft. wide, weighing 3 tons each, built up of long spiral galvanised steel plates 8 in. wide by 1·32 in. thick, and so arranged that there is a radial gap of $\frac{1}{8}$ in. between the plates through which the cooled air passes. Each air cooler has nine of these cylinders, taking 5400 ft. of plate (one for each of the first three stages and two for each of the remaining ones), the nine arranged to revolve on a common shaft at about 3 r.p.m. The lower part of each revolving cylinder is immersed in water of a lower temperature than the air, so that the air in passing the freshly wetted surface of the upper portions of the cylinders has its temperature reduced. The two water-tanks are piped up to the evaporators, of which there are six, each consisting of a cast-iron cylindrical tank 4 ft. 7 in. inside diam., and 12 ft. 3 in. high, and kept full of water to be cooled.

In them are immersed the coils of piping containing the ammonia undergoing evaporation. This piping is $\frac{15}{16}$ in. internal diam. and $\frac{3}{16}$ in. thick, there being over ten miles total length in use. The water in the evaporator is kept in motion by paddle stirrers working off bevel gearing and belting. There are six main ammonia compressors—one per stage—and each is a single cylinder double-acting engine of about 12 in. diam. by 21 in. stroke and belt driven at 80 r.p.m. from a 120 b.h.p. induction motor working at 365 r.p.m. The condensers are of the double type, consisting of 48 stands of 2 in. and $1\frac{1}{4}$ in. piping, each stand being eight pipes high and 20 ft. long. The initial ammonia charge is 10,000 lbs., recommended by the makers, Messrs J. & E. Hall, Ltd., Dartford.

Three-phase current is supplied at 15,000 volts from the local substation to the plant, where two 550-k.v.a. transformers are installed, which reduce the voltage to 380, at which pressure it is supplied to the various motors. There are twenty-one motors

installed, and when the whole plant is working the power input is about 700 E.H.P.


1919
1920
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NOV. DEC. JAN. FEB. MAR. APR. MAY JUN. JUL. AUG. SEP OCT NOV. DEC. JAN. FEB MAR. APR. MAY JUN. JUL. AUG. SEP. OCT. NOV. DEC. JAN. FEB. MAR. APR.
LEVEL NO. 24 (ROCK)
LEVEL NO. 21 (ROCK)
LEVEL NO. 20 (ROCK)
LEVEL NO. 18 (ROCK)
LEVEL NO. 14 (ROCK)
LEVEL NO. 12 (ROCK)
LEVEL NO. 16 (ROCK)
LEVEL NO. 10 (AIR)
LEVEL NO. 14 (ROCK)
LEVEL NO. 12 (ROCK)
LEVEL NO. 10 (AIR)
LEVEL NO. 8 (ROCK)
LEVEL NO. 13 (AIR)
ADIT LEVEL (ROCK)
LEVEL NO. 8 (AIR)
AVERAGE AIR-TEMPERATURE ON ENTERING MINE
TEMPERATURE, IN DEGREES FAHRENHEIT
TEMPERATURE, IN DEGREES FAHRENHEIT

FIG. 48.—Rate of alteration in downcast rock and air temperatures.

Fig. 48 shows the effect of the installation on the mine tempera-

ture over a period of thirty months, during the first thirteen of which the plant was not in regular operation. At the end of about one year of running the exit temperature was lowered to the figure for which the plant was designed. The depths of the levels are tremendous, as will be seen by figures 49, 50, and 51, which show very markedly the drop in dry-bulb temperature, in wet-bulb temperature, and moisture content respectively since the plant

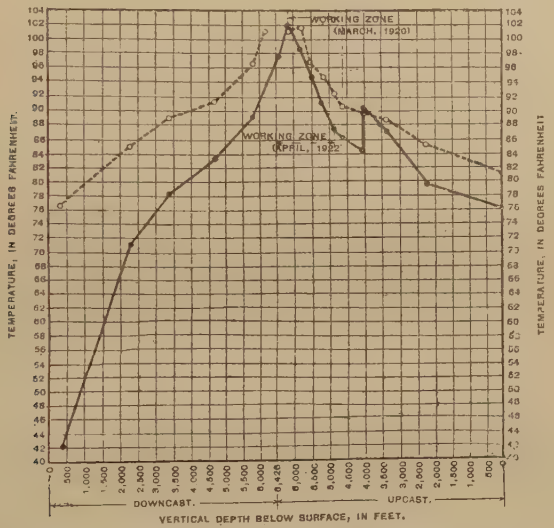

FIG. 49.—Comparison of typical former underground dry-bulb temperatures with those prevailing April 1922.

was installed. The other advantage arising out of the enterprise apart from the increased ventilating efficiency is the fall in the accident rate; the physiological conditions being much better the men are more alert, and can perceive and avoid danger more easily. From the same viewpoint it is reasonable to expect bigger outputs.

In this case the amount of air entering the mine is small, being only 80,000 cub. ft. per minute, because the cost of cooling great volumes of air is prohibitive, whilst in the City Deep case, cited above, the volume of air is over 400,000 cub. ft. per minute.

Another mechanical means of cooling the air is by the adiabatic expansion of compressed air, which will increase in cooling effect according as the pressure in the air lines is greater and the

expansion more nearly adiabatic. This method, owing to the

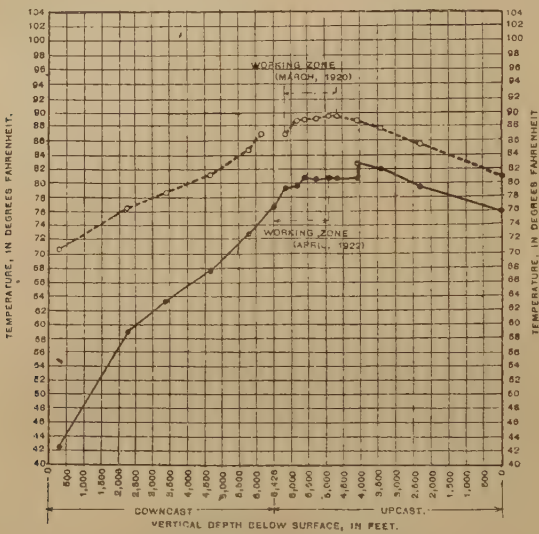

FIG. 50.—Comparison of former underground wet-bulb temperatures with those prevailing April 1922.

very low efficiency of compressed air in its production, transport, and use, will be very expensive, though it will be very easy to

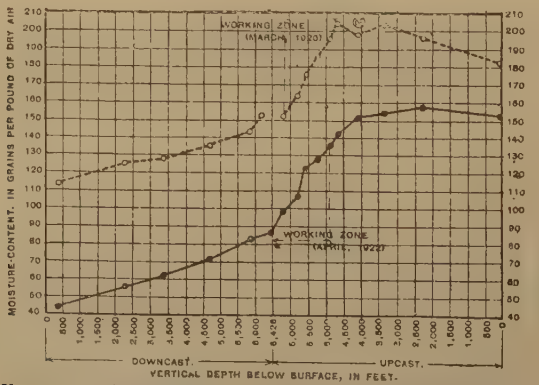
WORKING ZONE (MARCH, 1920)
WORKING ZONE (APRIL, 1922)
MOISTURE-CONTENT, IN GRAINS PER POUND OF DRY AIR
DOWNCAST
UPCAST.
VERTICAL DEPTH BELOW SURFACE, IN FEET.

FIG. 51.—Comparison of typical former underground moisture contents with those prevailing April 1922.

install, and will not need a cover over the downcast, as in the previous case.

Previous to discussing instruments used in the estimation of the amounts and effects of moisture on men and materials, we will briefly summarise the **Principles of Humidity.**

1. The temperature increases the ability of the air to absorb moisture.

2. The maximum absorption of moisture by the air is known as saturation, and its temperature point as the dew point.

3. The pressure due to the moisture depends on the temperature and degree of absorption.

4. The ratio of the amount of vapour present in the air to that needed to saturate it is called the **Relative Humidity,** *i.e.* the ratio of the actual to the saturation vapour pressure.

5. The weight of a cubic foot of water vapour at various pressures and temperatures is known as the **Absolute Humidity,** *i.e.* the amount present in unit space.

6. The weight and pressure of water vapour saturating a cubic foot of space is the same whether air occupies the space or not (see Dalton's Law of Partial Pressures).

7. All volatile liquids vaporise at all temperatures until the surrounding space is saturated.

8. When the surface-tension of a liquid overcomes the atmospheric pressure **boiling-point** is reached.

9. Aqueous vapour has a molecular weight of about 9, or sp. gr. of 0·312 (air = 1).

Storrow's Whirling Hygrometer.—Fig. 52 shows the original hygrometer designed by Mr J. T. Storrow, of Doncaster.* It consists of two accurate mercury thermometers, A and B, about 6 in. long, set in grooves, C, on cork pads in a wood frame, D, and fastened in with seccotine and two metal bars, E, E. The groove is cut through the wood for the greater part of the length of the tubes, enabling readings to be taken from either side and giving a better effect. The bulbs and reservoir, F, are also countersunk into the wood of the frame, and a protected grooveway is cut for the muslin, H, from the wet bulb, B, to the reservoir, J. The whole swings loose on the handle spindle, K, and can be easily whirled at a rate of 600 ft. per minute. It is carried in a small case, and it is more representative of the state of the air, owing to its motion, than the stationary types.

Notes on the Care and Use of Hygrometers.—1. Each thermometer should be calibrated accurately and checked against a

* J. P. Rees in *Third Report,* "Control of Atmospheric Conditions in Hot and Deep Mines."

standard thermometer of known accuracy over the range of temperature anticipated in its use.

2. Checking should be done with the bulbs of the standard and tested thermometers close together in the same medium.

3. The reservoirs should be filled with pure or distilled water, or clean rain water, because the deposit of solid filtered material from bad water impedes the transfusion of moisture.

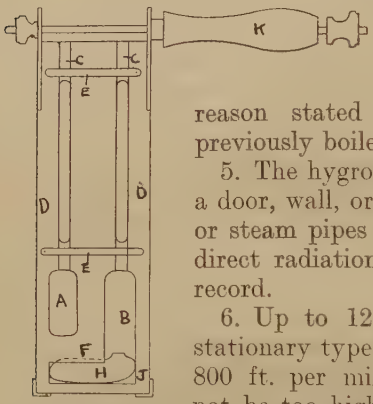

Fig. 52.

4. The muslin or silk on the wet bulb should be always in good condition, and should be changed monthly for the reason stated in 3. The thinnest muslin, previously boiled, should be used.

5. The hygrometer should not be placed near a door, wall, or other place where electric coils or steam pipes occur, as it is then subject to direct radiation and will not give a faithful record.

6. Up to 120° F. it should be, if of the stationary type, placed in a current of at least 800 ft. per minute, but the draughts should not be too high—this makes the reading too low. The air with this type should be fanned about in the region of the wet-bulb before reading.

7. The lowest reading at any time or condition is the one desired; therefore be sure when taking a reading to wipe away any moisture which may exist from the dry-bulb and take plenty of time.

8. In using the whirling type of hygrometer the instrument should be held out from the observer's side, and if used on the surface should be shaded from strong sunlight. Always take several readings and accept the lowest.

9. The whirling types, being more reliable than the stationary ones, are used to check the latter.

10. In very cold weather continued whirling is advisable so as to vaporise any ice on the muslin.

The moisture content of the atmosphere can be calculated by several methods, and many formulæ exist for finding the weight of moisture in the air and relative humidity. Most of these formulæ are empirical or involve the use of tables. Fig. 53 shows the hydrograph * prepared by Professor Briggs from

* Prof. Briggs, *Trans. Inst. M.E.*, xlv. 61.

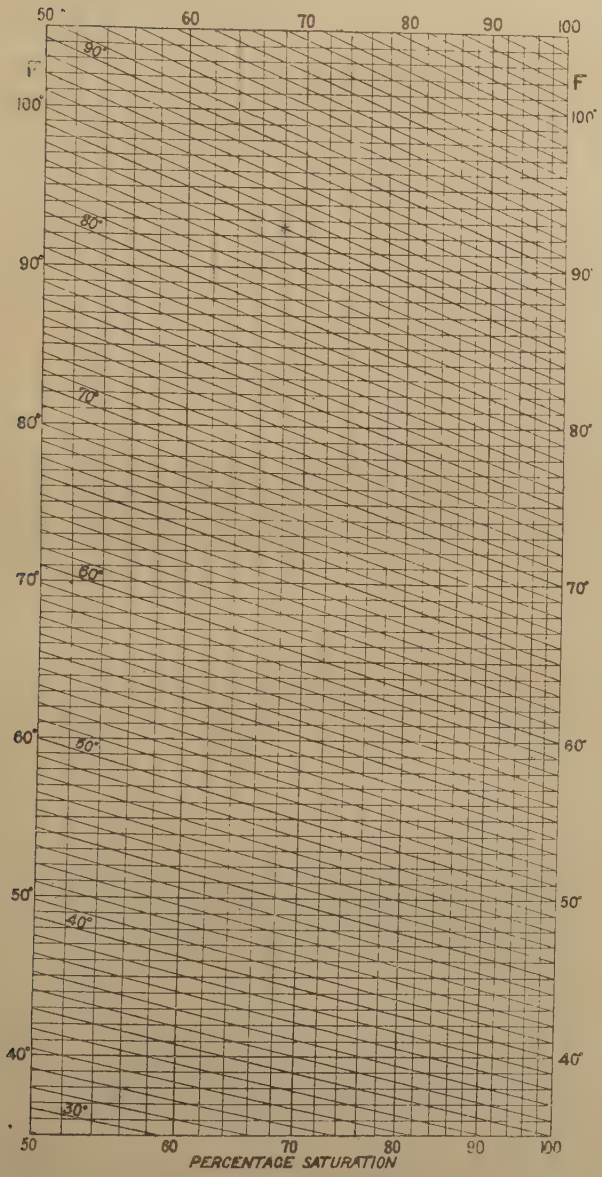

FIG. 53.—The Briggs hydrograph.

Glaisher's Tables as used by the British Meteorological Office. At the end of the book tables of important data relating to humidity, temperature, etc., are supplied, which are also arranged from Glaisher's Tables.

In fig. 53 the inclined lines are wet-bulb readings and the horizontal ones dry-bulb, the vertical lines being percentage saturation. When using the chart, start with the inclined line from the right corresponding to the wet-bulb reading; follow this line until the corresponding dry-bulb line is reached. From the point of intersection move by means of the vertical line to the top of the chart, and there read off the percentage saturation.

The range covered by the chart is 35° to 105° F. wet-bulb, and 50 to 100 per cent. saturation, which meets general conditions in British mines.

The psychrometric formula * of the Agricultural Department of the Weather Bureau, U.S.A., gives reliable results for the partial pressure in lbs. per square inch of the water vapour in the atmosphere. Thus where e is this partial pressure, P′ the corrected barometer in lbs. per square inch (*i.e.* the corrected barometer in inches of mercury divided by $\frac{29{\cdot}92}{14{\cdot}70}$ or 2·035), e' the saturation pressure of aqueous vapour at the wet-bulb reading t', and t the dry-bulb reading.

$$e = e' - 0{\cdot}000367\,\mathrm{P}'\,(t-t')\left\{1-\left(\frac{t'-32}{1571}\right)\right\} \quad . \quad (80)$$

P′, t, and t' are obtained from observation and e' from a temperature-pressure curve or chart for saturated steam or vapour. The dew-point temperature t'' in degrees F. can also be found from such a curve or chart being that temperature at which the pressure is e. From a saturated steam-temperature curve or chart the density Δ of the water vapour, in lbs. per cubic foot, is obtained, and consequently the vapour density Δ' at any actual temperature T will be obtained by Charles' Law as follows :—

$$\Delta' = \Delta\left(\frac{t''+459}{\mathrm{T}+459}\right) \quad . \quad . \quad . \quad (81)$$

In (56) the density of dry air is given as $\mathrm{D} = \dfrac{\mathrm{P}\times 144}{(459+t)\times 53{\cdot}34}$, where P is the corrected dry-air pressure and t the temperature,

* Prof. Ferrel in C. E. Lucke's *Engineering Thermodynamics* (Hill Publishing Co., London).

whilst the P′, in (80), is the pressure for the actual air under consideration. Now by Dalton's Law of partial pressures, since P′ is the sum of the partial pressures exerted by the dry air and the water vapour, then

$$P = P' - e \quad . \quad . \quad . \quad . \quad (82)$$

From these considerations the absolute moisture-content in grains of water vapour per lb. of dry air can be got thus $\frac{\Delta' \times 7000}{D}$, there being 7000 grains in a pound; should the unit preferred be grains per cubic foot at 32° F., and 14·7 lbs. per square inch abs., then since every pound of dry air under these conditions occupies 12·39 cub. ft., the expression for the moisture-content becomes

$$\frac{7000\ \Delta'}{12{\cdot}39D} * \quad . \quad . \quad . \quad . \quad (83)$$

In connection with measurements respecting humidity a very conspicuous place must be given to Dr Leonard Hill's kata-thermometer, which takes into account not only the temperature and humidity of the air, but its cooling effect, thus giving in numerical terms the physiological condition of the atmosphere. "It is undoubtedly the simplest ready means of indicating the cooling powers of an air current." †

Excessive cooling power can, in most cases, be adequately met by clothing, but no such simple remedy is available for defective cooling power, which, according to degree, has a pronounced effect on general health and the capacity for physical and mental work.

The factors which ordinarily influence the feelings of bodily comfort are— ‡

1. Temperature.
2. Air movement.
3. Radiant heat.
4. Humidity.

Ordinary wet- and dry-bulb thermometers give some measure of 1, 3, and 4, but these take no account of 2. Thus if a room with the windows closed had a fan working in it, the thermometer would give practically no indication of the fact, though the change would be very apparent to an observer in the room.

* Mr Eric Davies (Morro Velho, Brazil), *Trans. Inst. M.E.*, lxii. 333.

† Mr J. P. Rees, *Colliery Guardian*, 18/2/21.

‡ Mr B. W. Holman, "Ventilation and Working Efficiency, *Min. Mag.*, xxv., No. 6.

The kata-thermometer is a special instrument which measures directly the actual combined cooling effect, on a surface, of an air current of given velocity, temperature, and humidity; and also gives the cooling effect due to radiation, convection, and evaporation, without the use of anemometers and wet- and dry-bulb thermometers or other hygrometers.

It * weighs less and costs less than any reliable anemometer or hygrometer, and, above all, is extremely simple to manipulate and easy to read in a poor light. It (fig. 54) consists essentially of a large-bulbed alcohol thermometer designed for the ready measurement of its own rate of cooling from 100° F. to 95° F. This range of temperature is chosen to give a mean of 97·5° F. (36·5° C.), which is approximately that of the clothed body surface. A bulb about 4 cm. in length and 2 cm. in diameter, of known area, with a stem 20 cm. long, has been found to give a sufficient time-interval for accurate observation so long as the temperature of the air is not below 90° F. or over 105° F. The top of the stem terminates in a small reservoir which gives a margin for overheating the instrument without bursting the bulb and also allows the cooling to settle down to a steady rate before the 100° mark is reached. A thin muslin finger-stall is supplied with each instrument, so that either dry or ordinary wet kata readings can be taken.

Fig. 54.—Kata-thermometer.

The dry kata reading gives the cooling power by radiation and convection, while the wet kata gives the cooling power by radiation, convection, and evaporation—that is, the total cooling power of the ventilating current. The dry kata reading is sometimes used for quite a different purpose, namely, to measure the velocity of air currents when they are moving very slowly.

The Use of the Kata-thermometer.—To use the instrument for measuring purposes the bulb is heated in warm water at about 80° C. (this can be carried about in a thermos flask) until the spirit (coloured red) rises into the top reservoir. The column must be free from bubbles, and care must be taken not to let the top reservoir fill, otherwise the instrument may burst. The

* See B. W. Holman, "Ventilation and Working Efficiency, *Mining Mag.*, xxv., No. 6.

instrument is then allowed to cool, and the time in seconds taken in cooling from the 100° mark to the 95° mark is noted with a stop watch. The first reading should be neglected so as to allow the glass time to get into equilibrium. The mean of three or five readings may then be taken as desired. This mean time (T) divided into the factor (K) of the particular instrument gives the cooling power (H) directly in millicalories per square cm. of cooling surface per second. One millicalorie per square cm. per second is equal to 0·4 milli-B.Th.U. (thousandths of a British Thermal Unit) per square foot per second. Each instrument is carefully calibrated by the makers and the factor (K) engraved on the back of the stem. This factor is determined by observing the dry kata cooling power in still air at a standard temperature, t. When the temperatures are expressed in degrees centigrade, and 36·5 is the mean temperature of the kata during the observation time T, the factor is given by the empirical formula $K = 0{\cdot}27\ (36{\cdot}5 - t)\ T$. But we also have by definition $K = H_d\ T$, where H_d is the dry cooling power. If the difference between this mean temperature and the temperature of the working place be expressed by ϕ, we get the dry kata cooling power H in millicalories per square cm. per second, represented by the formula $H_d = 0{\cdot}27\ \phi$, where the air is still, but when it is moving Dr Leonard Hill found by experiment that $H_d = (0{\cdot}27t + 0{\cdot}49V^{0{\cdot}5})\phi$, where V is the velocity of the air current in metres per second (1 metre per second = 2·24 miles per hour). Hence if ϕ is known the velocity of the air current can be determined by taking the dry kata reading. This method is useful for measuring very low velocities, but breaks down with an alternating or badly eddying current. With low continuous currents an accuracy of within 10 per cent. error may be expected, as against within 80 to 100 per cent. with an anemometer. Experiments which agreed well with each other were made at the National Physical Laboratory in their 4 ft. and 7 ft. wind tunnels, and at Oxford with a rotating arm machine. These tests showed the relations which will be given later.

The Recording Kata-thermometer.—The instrument (fig. 55) consists of a large bulb alcohol thermometer,* the bulb being cylindrical in shape and of a standard size, length 4 cms., diameter 1·8 cms. Into the bulb is introduced a resistance wire, this wire forming one arm of a Wheatstone's bridge. The arrangement is such that sufficient current flows through the coil to raise the

* By Messrs The Cambridge and Paul Inst. Co.

temperature of the alcohol in the bulb to 36·5° C. (97·7° F.), when the bridge is balanced. If the bridge stays in this balanced condition, then the heat being supplied by the current in the coil

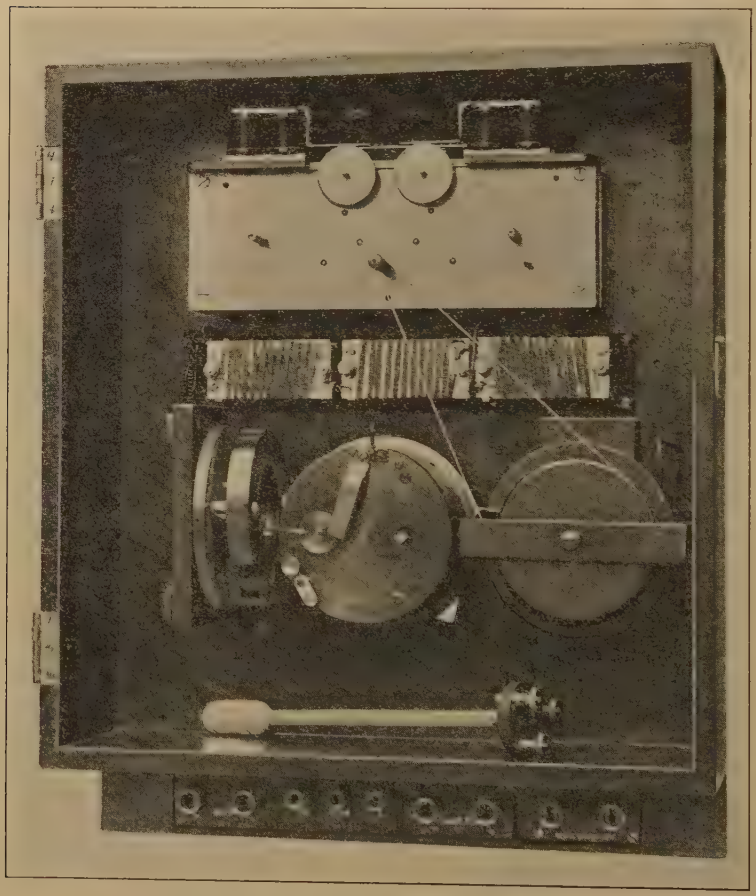

Fig. 55.—Recording kata-thermometer.

must be equal to that being lost by the bulb, and since the current through the coil is given by an ammeter in series with it, and the resistance of the coil is known, the amount of heat being produced, and therefore lost, per second is easily determined. If, now, the temperature of the surroundings fall, or more air movement occurs, the bulb will lose more heat, and if the current remains

constant its temperature will tend to fall. When this happens the resistance of the wire is reduced, the bridge balance upset, the galvanometer needle is deflected to one side and completes a circuit which causes a reduction in the resistance of the main circuit, which allows more current to flow through the bridge and so brings the temperature of the bulb back to 36·5° C., when the needle moves back to its balanced position, the ammeter indicating the increased current. If the conditions are such that

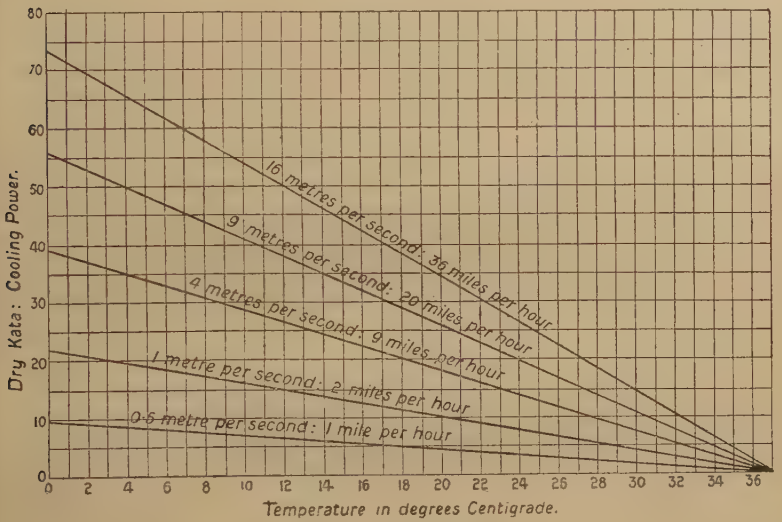
Dry Kata: Cooling Power.
Temperature in degrees Centigrade.
16 metres per second: 36 miles per hour
9 metres per second: 20 miles per hour
4 metres per second: 9 miles per hour
1 metre per second: 2 miles per hour
0·5 metre per second: 1 mile per hour
0
10
20
30
40
50
60
70
80
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32 34 36

FIG. 56.—Dry kata chart.

the bulb begins to lose less heat the temperature tends to rise, and the galvanometer needle is deflected in the opposite direction, in this case throwing in additional resistance so that the current is reduced.

It will be seen, therefore, that the ammeter gives the current required to keep the " kata " bulb at a constant temperature of 36·5° C., and that this supplies directly a means of knowing the heat lost from the bulb per second. The record is obtained by the ammeter needle being made to press at regular intervals on a piece of paper drawn beneath it, a typewriter ribbon being between the needle and paper. The recording kata allows continuous records to be taken, *e.g.* of the cooling power in the air of any place in a mine. By its means the cooling power at

any moment in any part of a mine might be made manifest to the manager in his office.*

The instrument is a modification of the ordinary kata-thermometer, and the heat loss, as given by the recorder, is expressed in the same units as those adopted for this instrument; that is to say, the number of millicalories (1 gm. calorie = 1000 millicalories) lost per second per square cm. of bulb surface, this being termed the cooling power "H." The range covered by the recording "kata" is from 2 millicalories per second per square cm. to 30 millicalories per second per square cm., *i.e.* cooling powers from 2 to 30. The cooling power of a comfortable room should be between 6 and 7 dry kata. The higher powers allowed for are to enable outdoor records to be obtained.

The relations between the heat loss H, wind velocity in metres per second v, and temperature in degrees centigrade t, are—

Where H_d is the dry and H_w the wet kata, t the dry- and t′ the wet-bulb reading centigrade.

For still air—

$$H_d = \cdot 27\ \phi \qquad (84)$$

$$\phi = 36{\cdot}5 - t, \text{ and } \phi' = 36{\cdot}5 - t' \qquad (85)$$

For wind velocities greater than 100 ft. per minute—

$$H_d = (0{\cdot}13 + 0{\cdot}47v)\ \phi \qquad (86)$$

$$H_w = (0{\cdot}10 + 1{\cdot}10\ V^{0.33})\ \phi' \qquad (87)$$

For wind velocities less than 100 ft. per minute—

$$H_d = (0{\cdot}22 + 0{\cdot}34v)\ \phi \qquad (88)$$

$$H_w = (0{\cdot}35 + 0{\cdot}84\ v^{0\ 33})\ \phi' \qquad (89)$$

These two latter formulæ expressed in feet per minute and degrees Fahrenheit become—

$$H_d = (0{\cdot}11 + 0{\cdot}016\ V^{0.33})\ \phi \qquad (90)$$

$$H_w = (0{\cdot}19 + 0{\cdot}08\ V^{0.33})\ \phi' \qquad (91)$$

It will be observed that with the aid of the above formula the wet kata cooling power can be calculated if the temperature, vapour tension, and velocity are known.

* Dr Leonard Hill, "Ventilation and Human Efficiency," *Inst. Min. and Met.*, 27/9/21.

TABLE XII.

Wet kata cooling power * in millicalories per sq. cm. of wet surface per second.	Remarks.
5	Extremely oppressive condition, inducing profuse perspiration ; there is very little evaporation from the respiratory tract, and a severe strain on the heat-regulating mechanism.
10	Oppressive.
15	Lower limit of satisfactory conditions, unless no physical work is being done.
20 and 25	Best conditions. The higher figure necessitating a fair amount of clothing or the performance of physical work.
30	Upper limit of satisfactory conditions ; induces a feeling of cold and danger of chill unless the body is well clothed or physical work is being done.
35	Uncomfortably cold, unless hard work is being done or winter clothing is worn.
40	Heavy winter clothing necessary.

This table is a rough approximation, and is subject to reservations, some of which may be mentioned here. For instance, if certain of the components, such as high air velocity or excessive dryness, are extreme, the conditions may be unhealthy, notwithstanding a satisfactory wet cooling power. Further, the amount and condition (*i.e.* wet or dry) of the clothing profoundly affect the cooling power of the air on the body.

The practical objections to the kata-thermometer in its present state of development are : (1) its frailness ; (2) its requirement of an external method of heating the fluid ; (3) the mixed units, *i.e.* Fahrenheit and metric ; (4) the limits, 100° F. or a little more, preclude it from very hot, deep mines. It will be seen that none of these objections are fundamental or irremediable.

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* E. H. Clifford, of the City Deep Mine, *Colliery Guardian*, 18/3/21.

CHAPTER V.

MINE PRESSURES AND RESISTANCES.

The idea of force as expressed by $\frac{Wv}{gt}$ conveys very little to the mind unless we state upon what area it acts. When the force per unit area is given the pressure is then defined. Assuming the weight of the air under ordinary conditions to be 0·0766 lb. per cubic foot, and the velocity v ft. per second, we find from (35) that it exerts an effort of

$$\frac{0{\cdot}0766\,v^2}{32{\cdot}2} \text{ or } 0{\cdot}0023\,v^2 \text{ ft.-lb.} \quad . \quad . \quad . \quad (92)$$

If the human body offers an area of, say, 8 sq. ft. of resistance, and the velocity of the air is 1800 ft. per minute, we get a total body pressure of—

$$\left(\frac{1800}{60}\right)^2 \times \frac{0{\cdot}0766}{32{\cdot}2} \times \frac{8}{1} = 17{\cdot}12 \text{ lbs.}$$

In a similar manner it can be shown that a man weighing 10 stone would be floored by a 60 m.p.h. gale. In this way the reader may be able to grasp the significance of ventilation pressure. The pressures in mine ventilation, are, like those in chimney draughts, town gas mains, etc., measured in inches of water column. A cubic inch of water exerts a pressure of $\frac{62{\cdot}3}{1728} = 0{\cdot}036$ lb. per sq. in., so that every inch of water column registered on a gauge by the ventilating pressure will equal $0{\cdot}036 \times 144 = 5{\cdot}2$ lbs. per sq. ft. Another way of attaining this result is to note that, to support an inch of water in an open gauge against the atmosphere, the same force per unit area is needed if the gauge is 1 sq. in. in cross-sectional area, 1 sq. ft., or any other area. One inch on a square foot is $\frac{1}{12}$th of a cubic foot, as shown by fig. 57, or $\frac{62{\cdot}3}{12} = 5{\cdot}2$ lbs. per sq. ft. approx. as before.

E.g. what is the pressure when a water gauge reads $3\frac{1}{2}$ in. ?

$3\frac{1}{2} \times 5{\cdot}2 = 18{\cdot}2$ lbs. per sq. ft.

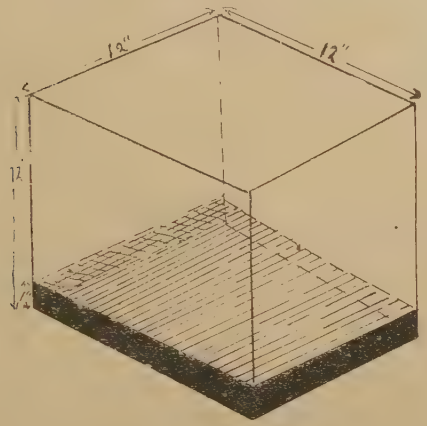

FIG. 57.

Now the pressure on each square inch of the base of a cubic foot of water is obviously $\frac{62{\cdot}3}{144} = 0{\cdot}434$ lbs. (see fig. 58), *i.e.* a

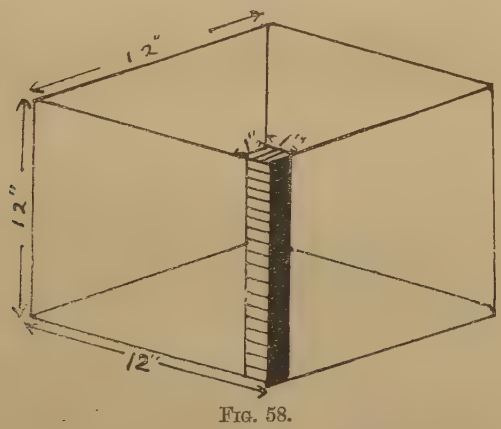

FIG. 58.

column of water 1 ft. high and 1 sq. in. in cross-section weighs or exerts a pressure of 0·434 lb. on the square inch base, or 2·3 ft. of water column exerts a pressure of 1 lb. per square inch. Thus the ordinary atmospheric pressure of 14·7 lbs. per square

inch will balance a column of water (surmounted by a vacuum) of $14{\cdot}7 \times 2{\cdot}3$, or nearly 34 ft. Water is 815 times as heavy as ordinary air; that is to say, 1 in. of water column will support 815 in. or about 68 ft. of air column. Stating this number generally for any densities, d for air and d_1 for water, at the same temperature and pressures it will be—

$$\frac{d_1}{12d} \text{ ft. of air column} = 1 \text{ in. W.G. (about 68 ft. normally)} \quad . \quad (93)$$

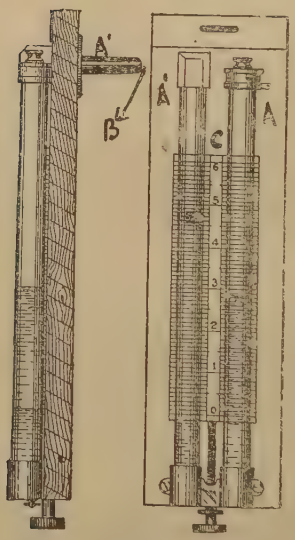

FIG. 59.—Ordinary water gauge. (Davies, Derby.)

This is sometimes written $\frac{1}{0{\cdot}015}$ and $\frac{d_1}{d}$ as $\frac{1000}{1{\cdot}2}$ for convenience.

Now it will be obvious that, other things being equal, the greater the difference between the intake and return air pressures the greater the water column difference between the two. The most common method of measuring this difference is by means of the water gauge, W.G., or by the Pitot tube. Fig. 59 shows a water gauge of the usual type to be seen at all collieries. It measures the difference in "static" pressure between the atmosphere and the return air, or between intake and return, *i.e.* no isolated reading is taken of the velocity head; this is measured by the Pitot tube. When the W.G. is spoken of in a general or unqualified way the above W.G. is implied. Such a gauge (fig. 59) consists of a glass U-tube A A′, one limb, A, open to the atmosphere or intake, the other one A′ connected to the return, its open mouth-edge B parallel to the direction of the current and not at right angles to it.

An adjustable sliding scale, C, graduated in inches and tenths, can be arranged to read the difference in the water levels due to the different pressures in the two limbs. Fig. 60 shows a continuously recording water gauge, a very valuable adjunct for any colliery, because it provides a very satisfactory check on the working of the ventilating plant, giving an automatic indication of the pressure conditions. The base of the apparatus is

formed of two cylinders of equal width which merge into one another at the extreme base, forming one large cylinder. Water is poured into the cylinders until the requisite point for the correct working level of the apparatus is reached. In each cylinder is a float, the two being joined together by a chain passing under a wheel in the single cylinder at the base. Fixed to one of the floats is a vertical rod *h*, to which is attached the recording pen *g*. The top of the apparatus consists of a glass case containing the drum *i*, on which the recording chart is placed, the drum being rotated about its vertical axis by a clock fixed on the top of the case. The clock is arranged to run for seven days or twenty-four hours as may be most expedient. The chart has both horizontal and vertical divisions, and so indicates the pressure and time simultaneously. The outside air is admitted through the inlet pipe C, into the chamber containing the float to which the pen is *not* attached, and bearing on the surface of the water every variation of pressure causes the float to rise or to fall, so causing the recording pen to give a faithful record. The recording float chamber is connected to the fan drift, return, or whatever air is being compared with that in the C limb. The aneroid principle is also applied to recording gauges.

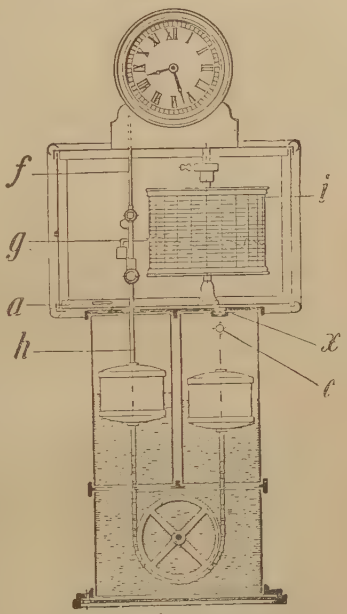

FIG. 60.—Recording water gauge. (Davies, Derby.)

It is well to note here that the water gauge does not register absolute atmospheric pressure, as does the barometer, but only the difference between it and that on the return side. This return side will exert less pressure than the atmosphere by reason of the air having had to expend some of its pressure in overcoming the various resistances in the mine in order to create ventilation, consequently the water in the return air limb will be higher than that on the intake or atmospheric side. The

difference, known as the W.G., is the measure of the ventilating effect.

E.g. 2 in. difference, W.G. $= 5{\cdot}2 \times 2 = 10{\cdot}4$ lbs. per square foot of ventilating pressure.

A barometer reading of 1 in. corresponds to $0{\cdot}49 \times 144 = 70$ lbs. per sq. ft.

A water gauge difference of 1 in. corresponds to $0{\cdot}036 \times 144 = 5{\cdot}2$ lbs. per sq. ft.

Water gauges are only used for comparative purposes; checking the daily working of the fan and condition of the mine.

If the weight of a cubic foot of air in a column be d lbs., and the height or head of the column be h ft., then the pressure p lbs. per sq. ft. exerted by the air is

$$p = dh, \text{ whence } h = \frac{p}{d} \text{ or } d = \frac{p}{h} \quad . \quad . \quad (94)$$

but W.G. $= \frac{p}{5{\cdot}2} = \frac{dh}{5{\cdot}2} = \frac{h}{68}$ approx. for normal atmospheric conditions of freezing-point temperature and pressure, *i.e.* dry air,

or by (52) $$= \frac{1{\cdot}3212\mathrm{B}h}{(459+\mathrm{T})5{\cdot}2} = \frac{0{\cdot}254\mathrm{B}h}{(459+\mathrm{T})} \quad . \quad . \quad . \quad (95)$$

and at ordinary pressures and temperatures

$$= \frac{0{\cdot}254 \times 30h}{(459+60)} = \frac{h}{68}.$$

Therefore for all cases the feet of air column equivalent to 1 in. W.G. should be deduced. In the absence of sufficient data 68 is often used.

We have seen (Chapter II.) that the head-producing flow or ventilation is called the motive column, M, so that it and h above are synonymous when h is measured as a result of comparing the mine return and the outside atmospheres, *e.g.* by the fan drift water gauge. This head being the mine resistance,

then by (59) $\mathrm{M} = \frac{\mathrm{D}(\mathrm{T}-t)}{459+t} = h = \frac{p}{d}$ where d is the density of the air.

From the above it will be seen that this cannot be stated, as is often done, as being exactly equal to 68 W.G., since it is the hotter and lighter or return air, and also contains the most moisture.

E.g. at a certain mine the shafts are 2200 ft. deep, and the

temperatures in the downcast and upcast 61° F. and 91° F. respectively, the barometer reading 30 in.

Deduce

1. The motive column.
2. The weight of a cubic foot of air.
3. The ventilating pressure.
4. The water gauge.

(1) $M = 2200\dfrac{(91-61)}{459-61} = h = 127$ ft. upcast air.

(2) $\dfrac{1{\cdot}3212\times 30}{459+91} = 0{\cdot}0720$ lb. per cub. ft.

(3) $p = hd = 127\times 0{\cdot}072 = 9{\cdot}144$ lbs. per sq. ft.

(4) $\dfrac{p}{5{\cdot}2} = \dfrac{9{\cdot}144}{5{\cdot}2} = 1{\cdot}76$ in. W.G.

The head created by the fan will be due to a force by (7) of $\dfrac{WV}{gt}$, where W is the total weight in lbs. of air being delivered by the fan in the time t seconds at the velocity of V ft. per second.

Therefore if W is the weight of air delivered per second, then the force supplied by the fan is $\dfrac{WV}{g}$ lbs. per second, and since it acts through a distance V ft. in one second the work done is $\dfrac{WV^2}{g}$ ft.-lbs. per second.

This is the same amount of work as a perfect fan would perform by moving the weight of air W through H_1 ft. in one second, *i.e.* $W\times H_1$ ft.-lbs.

$$\therefore\ WH_1 = \frac{WV^2}{g}$$

$$i.e.\quad H_1 = \frac{V^2}{g} \qquad . \qquad . \qquad . \qquad . \qquad (96)$$

Now this velocity head produced by an enclosed fan must be greater than that set up in the air in any part of the mine.

Similarly the head h between any intake and return in the mine will be less than the total head set up by the fan, since the latter has to overcome the friction of itself and of its ajutages. So the fan drift water gauge gives the truest reading of the mine

conditions though it reveals nothing direct regarding the ventilator resistance. The difference of pressure for an open fan is only $H_1 = \frac{V^2}{2g}$, or half that produced by the previous fan where a casing and expanding chimney are implied. The reason why this increase of pressure is set up by the enclosed fan is because of the restitution of energy stored up as velocity in the air leaving the fan tips, which is converted into pressure-head by increasing the area of the chimney, thus obeying (62). This subject we shall treat later.

The proper method of determining the ventilating pressure, on which depend the mechanical efficiency of the fan and many other radical ratios, is a matter of some doubt owing to the uncertain results obtained with the different forms of water gauge. The reading is always affected by the position and type of gauge, also by the velocity of the air current. It may vary with the ordinary type of gauge, shown in fig. 59, in different positions in the same part of a fan drift. The Pitot tube (fig. 62), gives no such variations in the same place if properly handled. Fig. 61 shows four forms of water gauge.

The usual form, No. 1, is not reliable except when the velocity is low, because by (62), when velocity increases due to a restriction of area the pressure decreases and, consequently, the height of the return limb will be greater. No. 2 has the same drawback. No. 3 shows an equalising box which reduces the velocity effect, and if only having a fine bore connection to the low-pressure side will register the exhaustion effect only. No. 4 is a correct form,* for here the velocity head is eliminated and only the static head is registered. To discuss the velocity head in more detail we will consider the water-gauge at the surface in the fan drift, which is its proper place for a complete mine check. The pressure in the drift will be less than that of the atmosphere owing to the weakening suffered by the air in overcoming the mine resistances. It will be seen at once that long and tortuous

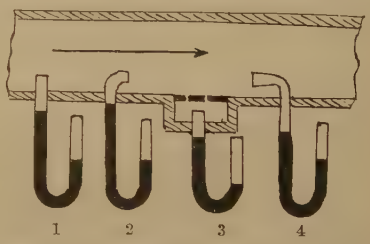

FIG. 61.—Different types of water-gauge tube-termination in a fan drift.

* However, see also Mr Thos. Bryson, "Wigan," *Trans. Inst. M.E.*, xlviii. 50.

Should read Total Head.

airways offering much rubbing surface to the air, and restricted airways or falls impeding the air, set up resistance which shows itself in increased difference on the gauge. On the other hand short and uniform roads show small difference, *e.g.* if the separation doors at the pit bottom be opened the whole of the air is short-circuited and passes straight up the upcast without encountering the resistance of the mine other than that of the shafts. The air will consequently lose very little pressure and the gauge show little difference. The normal difference in pressure is called the static pressure or head, or static water-gauge, *i.e.* it is the difference between two lots of quiescent air. Now the theorem of Bernouilli states that there is a loss of pressure consequent on increased velocity, and this can be stated thus—

A pound of air in the outside atmosphere possesses a certain amount of pressure energy, being still, but that in the drift is moving, therefore it possesses some kinetic energy. This K.E. will increase as the velocity increases. Since energy cannot be created or destroyed a fall in P.E. would occur as the K.E. got greater, and a gauge put in at this increased velocity point would mislead us by revealing a high head which we might be deceived into thinking a static head, whereas the increase in difference of the limbs is due to the above causes. We must, then, ascertain the effect due to velocity before we can correlate results with gauge readings in other parts or obtain a proper value for the exhaustion effect.

If we measure with a barometer the pressure b of the air in the fan drift and that of the atmosphere B outside, the head registered is $B - b$, but this is too great by an amount due to velocity, which we will call b_1. This makes the real reading $B - b - b_1$.

The student is warned to keep the meaning of the words *pressure in the air* and *water gauge* apart and distinct, even though they are different expressions for the same thing. We have already shown that if the pressure *of the air* in the return increases, the water-gauge limb readings approach nearer to each other, showing that the pressure of the air is getting nearer to that of the atmosphere, *i.e.* the greater the mine-air pressure the less the water-gauge difference.

The ventilating pressure set up by an exhausting fan is really the lack of pressure of the mine air due to the resistances it has to overcome, and the greater the resistance the greater the gauge reading or fall of pressure registered. In any case fall of mine air pressure means higher W.G. From this it will be seen that

if the gauge reading should be b_1 in. less there must be an equivalent pressure *in the air current*, but recorded by the gauge as a fall or difference; this is the true Venturi effect. (Imagine your arm thrust into the airway through a side hole, the air-pressure on the palm facing the current will tend to break the arm, but if the hand be taken out and placed over the hole, completely covering it, a suction effect will be experienced.) This pressure, b_1, can be ascertained by using two water gauges, one of type 1, fig. 61, with the open mouth-edge of its drift-end parallel to the direction of the current, and the other of type 4, with its open mouth facing the current. The air in the latter case will impinge on the gauge mouth and thereby lose its K.E., which is converted into an exactly equal amount of P.E., so that there will be a greater pressure in the tube than in the surrounding air by just this amount. This velocity head is the difference between the two gauges, and is always additive, because it increases with the velocity and vanishes when the velocity is zero. In other words, the *ordinary* water-gauge difference is always too great for a static reading (by b_1).

The increase of pressure in the air due to velocity, *i.e.* the dynamic head, is by definition and (4*a*)

$$h_v = \frac{V^2 d}{2g} \text{ lbs. per sq. ft.} \quad . \quad . \quad . \quad . \quad (97)$$

and by (93) $$W.G_v = \frac{12V^2 d}{2gd_1} \text{ in. of dynamical water gauge} \quad . \quad (98)$$

From this it will be seen that a dynamical water gauge of 1 in. is produced by a velocity of 4000 ft. per minute.

The real static pressure in inches is

$$B - b - b_1 = \text{ordinary water gauge} - W.G_v \text{ in.} \quad . \quad (99)$$

The dynamic gauge is often so small as to be considered negligible, hence the reason why the ordinary gauge is frequently called the static gauge.

Since the dynamic head represents work done by the fan it must be added to the static head to get the total work done.

Thus the air in motion in mines is influenced by

(1) The static pressure.
(2) The velocity or dynamic pressure.
(3) The combination of (1) and (2) or total pressure.

The maximum static water gauge will be obtained by noticing

the water gauge on the fan inlet, and that on the exhaust.* The difference between these is the maximum static water gauge.

The dynamic water gauge at the fan outlet will register the energy of the fan which is lost, and so the gauge should be placed on the outlet when deducing the effort of a fan; this is also the region of least eddies. If only a check on the velocity is needed the inlet side may be used provided the gauge is not too near the fan where eddies are likely to be present.

The dynamic head (98) is usually got by using a water gauge

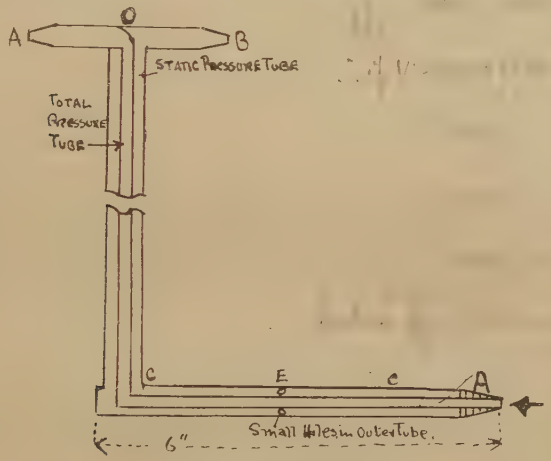

FIG. 62.—The Pitot tube.

one limb of which pierces the drift as in case 1, fig. 61, the other end also piercing the drift but having its mouth turned as in Case 4, same figure, there being no atmospheric limb.

For accurate work the Pitot tube, fig. 62, is used, and working on the above principle the static head is cancelled and only the dynamic head registered.

It consists of two concentric tubes containing air, the inner or dynamic tube A faces the current, whilst a few small holes E in the side of the outer tube C, $\frac{1}{16}$ in. diam., admit air for the static limb. The T-piece O is to facilitate connection to a water gauge. When leg A alone is connected by rubber tubing to the water gauge, the total pressure (static plus dynamic) is given. When only B is connected the static pressure only is given, and when both A and B are connected simultaneously the velocity or

* Mr D. E. Thomas, *Trans. Inst. M.E.*, l. 487.

dynamic pressure is given (see fig. 63). This pressure is by (97) proportional to the square of the velocity, or the velocity is proportional to the square root of the pressure. Thus if the dynamic water gauge be found to be 0·25 in. and the area 50 sq. ft., the quantity passing is

$$50 \times 4000 \times \sqrt{0{\cdot}25} = 100{,}000 \text{ cub. ft. per minute.}$$

Crude forms of the Pitot tube should be avoided, because only a skilled person exercising great care can operate the best improved forms: hence the reason why it is not more generally used. A correction or constant has to be applied to most Pitot tube readings, and this is derived in a similar manner to that

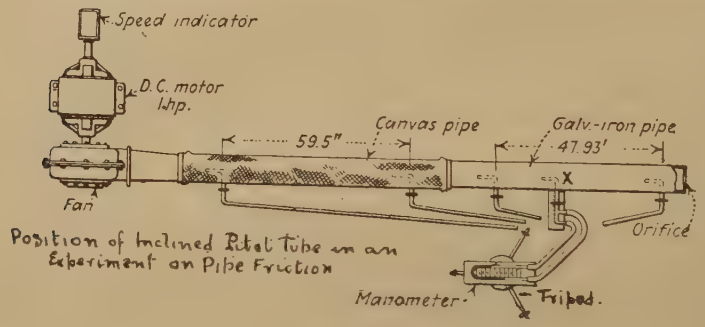

Fig. 63.

employed for the bracketed portion of (70). Furthermore, in all kinds of water gauge sluggishness occurs: for more accurate work a mobile fluid like alcohol should be used. Oscillations of the liquid may be damped by pouring down into the bottom of the U-tube base a few spoonfuls of glass beads or shot; they tend to subdue any surface agitation. When very refined observations are required by water gauge or Pitot tube an inclined U-tube (see figs. 63 and 64) must be used. In this case the *vertical* distance between the liquid surfaces must be read. A 20 : 1 inclined tube can be read to 0·0005 in. of water.*

The following discussion, strictly speaking, only applies to straight, radially bladed fans.

A mine fan operates by throwing off air at its periphery from its interior, thus creating at its centre a more or less empty space or partial vacuum; the greater air pressure in the inlet causes flow towards the low-pressure region, so creating the ventilating

* *Eng. Min. Journal Press*, No. 23, 1922.

current. This pressure difference is called the depression of the fan. It is convenient to express different pressures as water-column depressions. Hence there are four kinds of depression, *i.e.*

Theoretical depression, H_1.
Initial depression, H.
Effective depression, h.
Useless depression, h_0.

These terms apply to the usual type of ventilation in this country, *i.e.* exhaustive or due to the exhaustive action of a fan. Where the air is forced into the mine the principle would be compressive.

Theoretical Depression, H_1.—We have seen by (94) and (4*a*) that the head h_1 or motive column due to a pressure p_1 when the air density is d, can be expressed as

$$\frac{p_1}{d} = h_1 = \frac{V^2}{2g} \quad . \quad . \quad (100)$$

Daniel Murgue discovered that a perfect fan, *i.e.* one in which all the K.E. of the ejected air is converted into P.E., and no friction or other losses exist, would provide a motive column twice as great as that due to the tip velocity given by (100).

Since no fan is perfect, twice the value of $\frac{V^2}{2g}$ cannot be obtained, but most fans give a motive column in excess of $\frac{V^2}{2g}$. Thus with a peripheral speed of V ft. per second the theoretical depression will be by (96).

$$H_1 = \frac{V^2}{g} \quad . \quad . \quad . \quad (101)$$

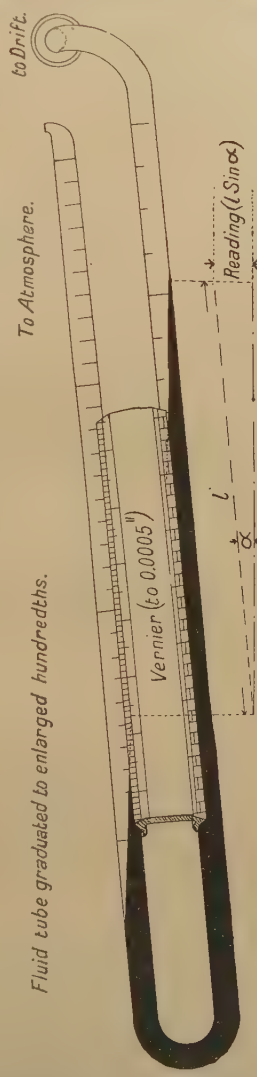

FIG. 64.—Inclined manometer.

This is the typical expression, but it is subject to much modification according to the design of the fan, see (247). (Only a radially bladed fan is being discussed here.)

The theoretical water gauge will be by (93) and (101).

$$\text{(Theoretical) W.G.} = \frac{12V^2d}{gd_1} = \frac{V^2d}{5{\cdot}2g} \quad . \quad . \quad . \quad (102)$$

It may also be conveniently expressed in terms of the speed of the fan, V ft. per second, due to N revolutions *per second*, and a diameter D ft. thus—$V^2 = (\pi DN)^2$.

$$\text{Substituting in (101) } H_1 = \frac{(\pi DN)^2}{g} = \frac{D^2N^2}{3} \text{ (approx.)} \quad . \quad (103)$$

$\therefore$ by (93)

$$\text{Theoretical W.G.} = \frac{\pi^2D^2N^2}{68g} = 0{\cdot}0045D^2N^2 \text{ (approx.)} \quad . \quad (104)$$

$$\text{and also theoretical W.G.} = 0{\cdot}00045V^2 \text{ (approx.)} \quad . \quad (105)$$

which results are useful in designing fans.

This value is never obtained in practice owing to the inability of the fan to discharge air into the atmosphere without velocity, and other causes.

Initial Depression, H.—This depression is not wisely named, a much better term would be the gross or total actual depression, since it results from the above. It is usually from 0·4 to 0·8 of the theoretical depression. This proportion of the initial to the theoretical depression is sometimes called the manometrical efficiency of the fan, K. A more correct definition of the manometrical efficiency will be found in the next chapter, section "Ventilation Efficiencies." Therefore the initial depression may be stated thus—

$$H = K\frac{V^2}{g} \quad . \quad . \quad . \quad . \quad (106)$$

$$\text{and the initial water gauge} = \frac{KV^2d}{5{\cdot}2g} \quad . \quad . \quad . \quad . \quad (107)$$

$$\text{or} \quad 0{\cdot}0045KD^2N^2 \text{ (approx.)} \quad . \quad . \quad . \quad (108)$$

$$= 0{\cdot}00045KV^2 \text{ (approx.)} \quad . \quad . \quad . \quad (109)$$

It cannot be got with the fan drift or mine water gauge under ordinary working conditions, as this only gives the mine "static" pressure.

To explain the "initial depression" more fully. If a fan running at uniform speed is exhausting from a closed chamber, eventually it will exhaust practically all the air from the chamber, and will spend its whole power to keep at bay the pressure of the

outer atmosphere. When this occurs, if a water-gauge be connected to the closed chamber and the outer atmosphere, it will register the " initial depression " of the fan at that speed.

This has been the subject of much controversy in the past owing to many experiments with curved blades, showing a slight rise in the water gauge (to begin with) as the mine inlet doors are opened gradually from the completely closed position. The water gauge should gradually fall as the inlet was opened, finally at full open showing the reading of the mine. Theoretically, it should not rise a little before falling. Many factors help to cause this slight discrepancy, chief among which are (1) the re-entry of air with certain types of fan blades; (2) the diminished density of the air in the closed drift; (3) leakage; (4) churning and eddying; and (5) greater virtual slip * of the air particles in fans having few blades, owing to there being less air in contact with the blades at the instant of discharge and so affecting the tangential velocity of the air, and, therefore, the depression.

Effective Depression, h.—This is the depression shown by the water gauge in the fan drift. It registers the resistance of the shafts and the mine and is that portion of the initial depression expended in overcoming this resistance.

by (94) $$h = \frac{p}{d}$$

(and effective W.G.) $$= \text{W.G.} = \frac{p}{5{\cdot}2} \quad . \quad . \quad . \quad (110)$$

The Useless Depression, h_0.—This is the portion of the initial depression not shown on the fan drift gauge, and is therefore that absorbed by the ventilator itself as a result of internal friction, etc.

$$\therefore h_0 = \text{H} - h \text{ or } \text{H} = h + h_0 \quad . \quad . \quad . \quad (111)$$
$$(\text{initial} = \text{total})$$

We cannot too often lay stress on the fact that this expression applies only to radially bladed fans; for curved blades it has no significance whatsoever, since many such fans have an observed or mine gauge in excess of the initial. Moreover, it is logically incorrect, because the left-hand or initial side is obtained by a fan working in a closed drift, *i.e.* taking no air, whilst the right-hand or total gauge side is got when the fan *is* taking air. The reason for its adoption here is because it provides the easiest means by which the starter

* Mr Jos. Parker.

can grasp the fundamental principles, and it also affords an opportunity to stress the above remarks.

In inches of water gauge

$$= \frac{KV^2d}{5{\cdot}2g} - \frac{p}{5{\cdot}2} \quad . \quad . \quad . \quad . \quad (112)$$

$$= \frac{KV^2d}{5{\cdot}2g} - \frac{dh}{5{\cdot}2} \quad . \quad . \quad . \quad . \quad (113)$$

$$= \frac{d}{5{\cdot}2}\left(\frac{KV^2}{g} - h\right) \quad . \quad . \quad . \quad (114)$$

or approximately

$$= \frac{1}{68}\left(\frac{KV^2}{g} - h\right) \quad . \quad . \quad . \quad (115)$$

The initial gauge is sometimes called the " total observed gauge," and by other terms which need consideration before use.

The Equivalent Orifice.—The resistance of a mine can be assimilated to the resistance set up by a thin plate with an aperture in it. The body of air flowing through such an aperture, like all other fluids under similar conditions, contracts into a waist, the cross-sectional area of which is given by Murgue as 0·65 times the area of the orifice. This contracted vein is known as the " vena contracta."

This opening in a thin plate, which in passing a certain quantity of air registers a water gauge, equal to that produced by the same quantity of air passing through the workings of the mine, is termed the " equivalent orifice " of the mine.

Again, note that the following remarks, allied as they are to the conceptions upon which the preceding formulæ are built, will, of necessity, be subject to the same very fine limitations. Consequently, for the average ventilator in practice the equivalent orifice notion is very likely to be erroneous ; but a knowledge of it forms one of the simplest means of comprehending the effect of different kinds and combinations of resistances.

The equivalent orifice is usually expressed in the following ways :—

(*a*) By Professor Rateau *—

$$a = \frac{Q}{\sqrt{gh}} \quad . \quad . \quad (116)$$

Where
a = orifice area in sq. ft.
Q = quantity in cub. ft. per second.

* See also Innes, *The Fan*, chap. iii.

(*b*) By Dl. Murgue—

$$a = \frac{Q}{0{\cdot}65\sqrt{2gh}} \quad . \quad (117)$$

g = acceleration due to gravity 32·2 ft. per second^2.
h = head of air column in feet.

Expressing (116) in the same form as (117) the vena contracta coefficient is found to be $\frac{\sqrt{2}}{2}$ or 0·707. In the majority of experiments which have been carried out on the equivalent orifice the results appear to prove that Rateau's formula is the most reliable, and that Murgue's figure, 0·65, is too low, and that it should be at least 0·7. Murgue's coefficient is usually adopted. This figure, although called a constant, is not absolutely constant, as is generally supposed, but varies somewhat under varying conditions. Mr T. Bryson, of the Royal Technical College, Glasgow, considers, as the result of much experimental investigation, that this figure is influenced by the relation existing between the area and perimeter of the orifice. This is a sound conclusion, since the passage of a fluid over a notched weir is influenced in the same manner and the conditions are dynamically similar in many respects.

From first principles the equivalent orifice formula is deduced as follows :—

By (25)	$a = \frac{Q}{v}$	Where Q = quantity passing in cub. ft. per sec.
„ Murgue	$= \frac{Q}{0{\cdot}65\ v}$	h = head of air pressure (in ft.) from gauge.
„ (6*a*)	$= \frac{Q}{0{\cdot}65\sqrt{2gh}}$	a = area of the orifice.
„ (93)	$= \frac{Q}{0{\cdot}65\sqrt{64{\cdot}4 \times 68 \text{ W.G.}}}$	v = velocity in ft. per sec.
„ taking quantity per minute	$= \frac{Q}{0{\cdot}65 \times 60\sqrt{64{\cdot}4 \times 68 \text{ W.G.}}}$ approximately	0·65 = vena contracta "constant," by Murgue.
„ clearing the denominator	$= \frac{{\cdot}000388\ Q}{\sqrt{\text{W.G.}}}$	
„ taking Q in thousand of cub. ft. per minute	$a = \frac{0{\cdot}388\ Q}{\sqrt{\text{W.G.}}}$ . . . . . (118)	

The above coefficient 0 388 becomes 0·356 in (116) when using Rateau's formula. (Note the approximation due to assuming $h' = 68$ W.G.")

The inherent head loss, h_o ft., due to the ventilator itself, can be expressed in inches of water gauge, $W.G_o$, by dividing by 68, or, more exactly, by (93), so we may now deduce an expression for the equivalent orifice, o, of the fan in terms of h_o or $W.G_o$, just as we did for the equivalent orifice of the mine. This is known as the **orifice or passage** of the fan and, as in the mine simile, its size is a measure of the resistance. The smaller it becomes the higher the ventilator's resistance, and *vice versa*.

Thus

$$o = \frac{Q}{0{\cdot}65\sqrt{2g(H-h)}} \text{ or } \frac{Q}{0{\cdot}65\sqrt{2gh_o}} \text{ by Murgue} \quad . \quad (119)$$

$$o = \frac{Q}{\sqrt{gh_o}} \text{ by Rateau} \quad . \quad . \quad . \quad . \quad . \quad (120)$$

Q being in cub. ft. per second, h_o in ft., and o in sq. ft.

$$\therefore\ o = \frac{0{\cdot}388Q}{\sqrt{W.G_o}} \text{ or } \frac{0{\cdot}356Q}{\sqrt{W.G_o}}$$

where the same units and procedure apply as to those of (118).

This internal resistance of the fan $W.G_o$ is not visible on the ordinary water gauge in the drift, but can be got by deducting from the water gauge in the closed drift the water gauge of the mine.

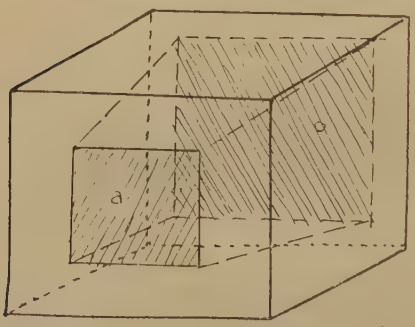

Fig. 65.

The fan resistance is usually less than that of the mine, and since it varies inversely as the orifice o, this fan orifice is generally greater in area than that of the mine a.

This is shown graphically in fig. 65. The relationship between a and o is very important because the former registers useful work done in overcoming the mine resistances, whilst the latter is a measure of the waste of the fan. It will be seen that a and o increase as the resistance decreases, and that o is lessened as the complications, friction, mass, and other resistances of the fan become greater. There is a limit, then, to the size as well as the

design of the fan. The orifice of the mine should be plotted regularly to give a check and comparison.

E.g. 1. In a mine having an equivalent orifice of 24 sq. ft. the W.G. is $2\frac{1}{4}$. What is the effect on the equivalent orifice if the W.G. is increased to 4 in. ?

$$a_1 : a_2 : \frac{1}{\sqrt{\text{W.G}_1}} : \frac{1}{\sqrt{\text{W.G}_2}}$$

$$24 : a_2 : \frac{1}{\sqrt{2{\cdot}25}} : \frac{1}{\sqrt{4}}$$

$$\frac{24}{\sqrt{4}} = \frac{a_2}{\sqrt{2{\cdot}25}}$$

$$a_2 = 18 \text{ sq. ft.}$$

E.g. 2. The orifice of passage of a fan is 80 sq. ft. What is the effect of doubling the internal resistance x ?

$$o_1 : \frac{1}{\sqrt{x}} : o_2 : \frac{1}{\sqrt{2x}}$$

$$\frac{o_1}{\sqrt{2x}} = \frac{o_2}{\sqrt{x}}$$

$$o_2 = \frac{o_1 \times \sqrt{x}}{\sqrt{2x}} = \frac{o_1}{\sqrt{2}}$$

$$= \frac{80}{\sqrt{2}} = 56{\cdot}56 \text{ sq. ft.}$$

If a and o be equal then, apart from anything else, at least 50 per cent. of the total ventilating pressure is spent on overcoming the resistance of the fan, which, if we consider other additional causes of loss, will be a greater proportion still of the total ventilating pressure. When $o = 2a$ the velocity in o will bear the same ratio to that in a as 1 : 2, but the pressures will be in the proportion as 1 : 4, since pressure varies as the square of the velocity, see (62). Again, if $o = 3a$ the velocities are in the ratio 1 : 3, and the pressures 1 : 9 respectively, or only $\frac{1}{10}$th of the ventilating pressure is used up in overcoming the fan resistance, the remaining $\frac{9}{10}$ths being usefully employed on the mine. Such a ventilator would be theoretically 90 per cent. efficient, but actually it is only (K × 90) per cent. efficient where K is the manometrical efficiency. Hence the reason why we place 0 8 H_1

as the highest value of H; this is really due to an actual efficiency of about 90 per cent. of the theoretical efficiency, which is the practical extreme.

It would appear from the above that the greater the ratio of o to a the better, but it will be seen that to decrease $W.G_o$ for the same conditions will need a remodelled, bigger, or faster-running fan perhaps. In any case, whether the size, mass, or speed be increased the machine friction is also increased, so the gain due to the increased ratio of o to a will be sooner or later outweighed by cost, power, etc. This occurs when o becomes greater than $3a$,* the best relationship of o to a being

$$o = 2 \text{ to } 3 \text{ times } a.$$

The Combined Orifice of Fan and Mine.—If the effective depression head h passes a quantity of air, Q, through the mine, then the unregistered depression h_o passes the same quantity also through the fan, so that we may express the combined orifice of fan and mine, A, thus (still considering a radially bladed fan), by Rateau—

$$A = \frac{Q}{\sqrt{g(h+h_o)}} \text{ or } \frac{0{\cdot}356Q}{\sqrt{W.G + W.G_o}} \quad . \quad . \quad (121)$$

where Q is in cub. ft. per second in the first case, and in thousands of cub. ft. per minute in the second. The quantity under the radical in the latter case, which we will call W for convenience, is the initial water gauge due to H.

i.e. $$A = \frac{0{\cdot}356Q}{\sqrt{W}}, \text{ or } \frac{0{\cdot}388Q}{\sqrt{W}} \text{ (by Murgue).}$$

Now since in this case $H = h + h_o$

and neglecting the common g, $a = \frac{Q}{\sqrt{h}}$ and $o = \frac{Q}{\sqrt{h_o}}$

then $$\frac{h}{h_o} = \frac{o^2}{a^2}$$

$\therefore$ combined orifice = $$A = \frac{Q}{\sqrt{H}} = \frac{Q}{\sqrt{h+h_o}}$$

* D. M. Mowat, *Trans. Inst. M.E.*, xliv. 96.

$\therefore$ by squaring $$A^2 = \frac{Q^2}{h+h_o} \quad . \quad . \quad . \quad . \quad (122)$$

and Q being constant $$A^2 \quad \frac{Q^2}{\frac{Q^2}{a^2}+\frac{Q^2}{o^2}} - \frac{1}{\frac{1}{a^2}+\frac{1}{o^2}} = \frac{a^2o^2}{a^2+o^2}.$$

Whence $$A = \frac{ao}{\sqrt{a^2+o^2}} \quad . \quad . \quad . \quad (123)$$

For any number of orifices a, b, c, d, etc., we can find their combined orifice by retracing (123) and obtaining the form

$$\frac{1}{A^2} = \frac{1}{a^2}+\frac{1}{b^2}+\frac{1}{c^2}+\frac{1}{d^2}+ \ . \ . \ . \text{ etc.}$$

or $$A = \sqrt{\frac{1}{\frac{1}{a^2}+\frac{1}{b^2}+\frac{1}{c^2}+\frac{1}{d^2}}} \ . \ . \ . \text{ etc.} \quad . \quad (124)$$

This expression is very significant, because it symbolises the important law, viz. : " the combined orifice due to fan and mine, or any number of other orifices, is equal to the sum of the squares of the reciprocals of such orifices." This is the usual method of stating the combined orifice law, but is scarcely a correct method. The use of this law in the design and performance of fans will be appreciated when we consider that the manometrical efficiency which shows how far a fan falls short of perfection (since no fan discharges at zero velocity into the atmosphere) depends largely on the combined orifices at the fan itself. Calling the inlet area i, and the chimney area c, and using any vena contracta coefficient k, we find by (123) and (119) that the orifice of passage o is

$$o = \frac{K}{k} \times \frac{ci}{\sqrt{c^2+i^2}} \quad . \quad . \quad . \quad (125)$$

also $$\frac{o}{i} = \frac{K}{k} \frac{\left(\frac{c}{i}\right)}{\sqrt{\left(\frac{c}{i}\right)^2+1}} \quad . \quad . \quad . \quad (126)$$

From this it will be seen, by using the values of k given previously, that $\frac{K}{k}$ varies from a maximum of 1·23 or 1·13 to a

minimum of 0·61 or 0·56, according as to whether we use the k of Murgue or Rateau. For modern fans this ratio of the manometrical efficiency to the vena contracta might be taken as unity. The ratio $\left(\frac{o}{i}\right)$ of the area of the orifice of passage to that of the inlet varies from 1·13 to 0·3, and is taken as an average of 0·65 by Peele.* (See note on this subject in following chapter, section "Fan Speeds.")

The orifice of passage and the mine equivalent orifice form very simple and convenient methods of comparing the resistance of fan and mine under different conditions, provided that we are dealing with straight, radially bladed ventilators. This method of comparison is by no means simple, if at all possible, when comparing differently bladed fans under different conditions. Many modern fans possess blades inclined forward in the direction of rotation, and most ventilating machines of this class tend, among other causes, to destroy their own vacuum by causing re-entry of air at the tips when being tested for initial depression with closed drifts. In this way they may well produce a higher registered effective depression than initial. This would cause a negative orifice of passage, which is absurd, or what is equally ludicrous, a manometrical efficiency of over 100 per cent. The orifice methods will be equally inappropriate when applied to backward bladed fans, thus restricting their use to the limits above mentioned. Professor Briggs † has suggested that the orifice method of calculating ventilation resistance be relegated to oblivion, and a unit adopted on radical scientific lines as is done in electricity. In this way the name of some eminent mining engineer might be perpetuated. Mr Penman ‡ suggests that the unit of mine resistance, 1 Garforth, be that resistance against which 1 in. of water gauge will cause unit quantity of air to flow, say 100,000 cub. ft. per minute.§ Thus it would be possible to compare the performance of any number or types of fans and mines just as any number of varied boilers, under different working conditions, may be compared by computing for each the equivalent evaporation at, and from, 212° F.

Furthermore, the orifice of a mine is not constant, not even from day to day, seeing that it depends upon the height of an

* Peele's *Handbook*, p. 1048.
† *Trans. Inst. M.E.*, 1922, p. 58.
‡ *Ibid.*
§ See also suggestion by Mr Joseph Parker in final chapter.

air column, which varies directly as the temperature, neither is it bound to decrease with the development of the mine; at Bentley Colliery it actually increases. It is obvious also that the effect of temperature can, in more or less degree, counteract the effect of increased resistance. Thus by clearing all cancellable fractions and the root sign from (116) or (117) we obtain $\frac{Q^2}{h}$, which forms a very convenient indication of the conductance, C, of the mine, *i.e.* the efficiency respecting its power to conduct a given quantity of air.

The reverse of this is the resistance R, and

$$R = \frac{h}{Q^2} = \frac{1}{C} \quad . \quad . \quad . \quad . \quad (127)$$

This forms the most rational and practical method of determining the suitability of a fan to a mine. Therefore by adopting the above unit of Professor Briggs and Mr Penman we may compare the ventilating accomplishment of any mines as follows: *e.g.* three mines, A, B, and C, taking 50,000, 150,000, and 300,000 cub. ft. of air per minute at pressures of $1\frac{1}{2}$ in., $2\frac{1}{2}$ in., and 3 in. W.G. respectively.

Express their resistances in the units discussed above.

$$A. = \frac{1\frac{1}{2}}{(0{\cdot}5)^2} = 6.$$

$$B. = \frac{2\frac{1}{2}}{(1{\cdot}5)^2} = 1{\cdot}11.$$

$$C. = \frac{3}{(3)^2} = 0{\cdot}33.$$

This method of measuring resistances has, it will be seen, the advantage of directness over the orifice method of inverse ratios, it is also unencumbered by any doubtful, or at least variable, coefficient. Adhering to the given signs and calling the fan resistance r, the expression (127) can be furthered thus—

$$Q^2 = \frac{H}{R+r} = \frac{h}{R} = \frac{h_0}{r} \quad . \quad . \quad . \quad (128)$$

TABLE XIII.—*Comparison of the Numerical Values of the Equivalent Orifices and the Corresponding Resistances in the Unit of Resistance proposed in the Text.*

Equivalent orifice in square feet.	Units of resistance.	Equivalent orifice in square feet.	Units of resistance.
5	60·80	55	0·52
10	15·20	60	0·42
15	8·65	65	0·36
20	3·80	70	0·31
25	2·43	75	0·27
30	1·69	80	0·24
35	1·24	85	0·21
40	0·95	90	0·19
45	0·75	95	0·17
50	0·60	100	0·15

Resistance.—From the previous remarks regarding pressure it will be clear that resistance and pressure are equal and opposite. Resistance is always referred to in terms of the rubbing surface, *i.e.* the surface of coal, strata, etc., against which the air rubs in the mine, and is practically the length of the airways traversed multiplied by the perimeter. It is usually, but not necessarily,

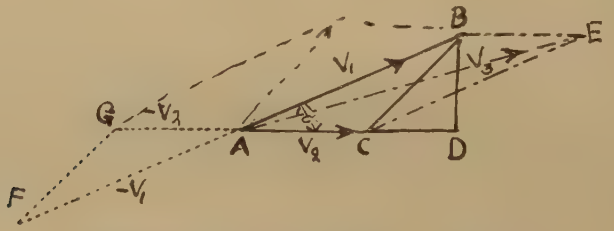

FIG. 66.—Loss of head due to change of direction.

distinguished from friction, which is often taken as the rubbing surface per square foot of section or $\frac{\text{rubbing surface}}{\text{area}}$. It will be seen later that velocity varies inversely as the square root of the rubbing surface, and, consequently, directly as the square root of the pressure. The total factors producing resistance are—

(i) The total superficial extent of all exposed surfaces in the mine—this is the chief cause of resistance.

(ii) Bends; this may be shown as follows: consider fig. 66,

which shows the change of direction of a current from A B, where the velocity is V_1, to A C, where, as a result of the deflection through the angle α, it will be less or V_2. The resultant velocity will be (to the same scale) the diagonal of the parallelogram, *i.e.* A E or V_3. But V_3 is the additive effect of the two vectors of velocity V_1 and V_2. If we reverse the vectors, as shown by the dotted lines, it will be clear that F G = B C is the difference of the vectors, and gives us graphically the loss in velocity, or $V_1 - V_2$. Now having the angle α and the two sides containing it, the remaining side of the triangle is

$$BC^2 = V_1^2 + V_2^2 - 2V_1V_2 \cos \alpha = \text{loss of velocity squared.}$$

By (4*a*) pressure loss (resistance increase) $= \dfrac{BC^2}{2g}$

$$= \frac{V_1^2 + V_2^2 - 2V_1V_2 \cos \alpha}{2g} \quad . \quad . \quad (129)$$

(iii) Variations in the sectional area. These cause eddying and rebounding if great and sudden. They make considerable demand upon the pressure.

(iv) Cages, trams, and other objects in motion.

(v) Regulators and excessive splitting.

(vi) Temperature changes and blasting.

(vii) The concussion of the air columns at mid-shaft during winding.

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CHAPTER VI.

THE PRODUCTION OF VENTILATION IN MINING.

VENTILATING currents may be created in the following ways, viz.—

(*a*) By difference of densities due to seasonal changes of temperature, etc., *i.e.* natural ventilation.

(*b*) By the entraining action of falling water; the water jet or the steam jet.

(*c*) By the air density differences set up by the temperature due to a furnace.

(*d*) By centrifugal machines or fans.

(*a*) **Natural Ventilation.**—When the tops of two shafts, A B and C D, fig. 67, are at different elevations there will be a difference also of temperature in each shaft. For this reason, in cold weather, the column E A will be heavier than column C F, with the result that a current will flow in the direction of the feathered arrow, so that D C becomes the upcast. Should the outside temperature exceed that of the mine, as in summer, the conditions are reversed. It will be seen that, in the spring and autumn, and in tropical climates between night and day when the current reverses due to the above conditions changing, the current is either nil or very sluggish. It is not intended here to convey the notion that natural ventilation is negligible;

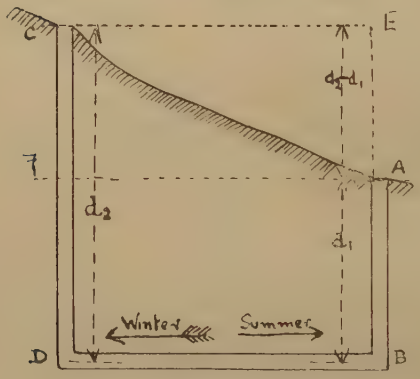


FIG. 67.—Natural ventilation.

its effect will be felt in any mine whether ventilated by fan, furnace, or otherwise. This effect is given * as—

$$Q = \sqrt{Q_1^2 + Q_2^2} \quad . \quad . \quad . \quad (130)$$

Where Q is the total quantity passing in the mine.
,, Q_1 ,, ,, produced by the fan.
,, Q_2 ,, ,, ,, ,, natural ventilation.

From which it will be seen that the individual effects of natural and fan ventilation are not together equal to their combined effect. Referring to fig. 67, it must be noticed that when speaking of a " column of air " its top is that of the upcast and its base that of the downcast, whichever of the shafts are performing these functions. The pressure p, producing ventilation, will depend on the difference of densities in these shafts, and if T_1, T_2, and T are the respective absolute temperatures of the air in downcast, upcast, and on the surface, and B the barometer reading in inches, then by (52)

$$p = 1{\cdot}3213B\left(\frac{AB}{T_1} + \frac{EA}{T} - \frac{CD}{T_2}\right) \quad . \quad . \quad (131)$$

Now in summer the conditions of fig. 67 will be such that the third term inside the bracket will be greater than the other two, p will be negative, and reversal will occur. When there is no difference of elevation, as in most English mines, (131) is much simplified to

$$p = 1{\cdot}3212 \times BD\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad . \quad . \quad (132)$$

where D is the depth, B the barometer in inches, and T_1 and T_2 the absolute downcast and upcast temperatures respectively.

E.g. at a mine 1000 ft. deep the upcast temperature is 90° F. and downcast 60° F.; the barometer stands at 30 in. What is the ventilating pressure due to natural ventilation ?

$$p = 1{\cdot}3212 \times 30 \times 1000\left(\frac{1}{519} - \frac{1}{549}\right)$$
$$= 4{\cdot}2 \text{ lbs.}$$

In addition to temperature changes and elevation, humidity will also affect natural ventilation since it tends to lessen the density of the air. The same remarks apply to bodies of water and other humidity influences in and about mines.

* *N.E. and Mid. Inst. Rept. on Mechanical Ventilation.*

(*b*) **The Entraining Action of Falling Water or Jets.**—This antiquated system, like the preceding one, is never relied upon alone to produce ventilation in modern British mines. It is usually adopted in small metalliferous mines possessing bodies of falling water, and in such cases its effect is often augmented by the exhaust air from power drills. Its action depends on a vertical column of water-pipes (fig. 68), with perforations at the surface and exit holes at any intermediate levels. The water in falling entrains the air, causing a suction action through the holes. The air travels down with the water, some of it escaping at the intermediate landing holes (if any), the main body descending to the sump. Here a flat or slightly conical dash-board breaks the column of water into a fine shower, the air being dispersed in all directions horizontally. The comparatively rare circumstances for its adoption are elevated shafts with gravity adits, otherwise there is considerable pumping expense.

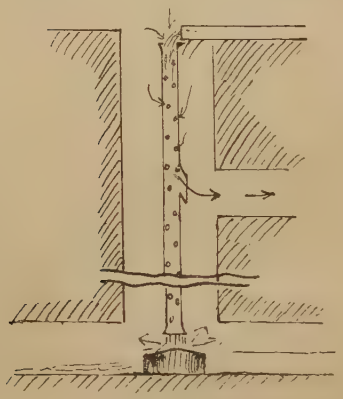


Fig. 68.

Jets.—Some of these operate on the injector principle, which is simply a modification of the above water-trompe principle. Steam jets used to be installed in non-winding upcasts, and the Americans still use them, especially in finished boreholes. In places where compressed air is installed, narrow heads and single entries are sometimes ventilated by a compressed air-jet as in fig. 69. The system is too wasteful and inefficient to dwell long upon, so further adaptations will not be dealt with here—it gives less than 10 per cent. of a fan effect.

(*c*) **The Furnace.**—In order to supplement the result of natural ventilation the furnace was introduced as an artificial means of producing a similar but magnified effect. It is placed at the base of the upcast shaft, and by its temperature rarefies the air there so that that of higher density in the downcast in tending to restore equilibrium causes a current. By this means the sluggish currents of the spring and fall, or between day and night in the tropics, are obviated. It is passing quickly from the mining industry owing to its wastefulness, inefficiency, and its

liability to ignite inflammable gases. Its continued adoption in some mines is due to the risk of excessive shrinkage of the strata, which would ensue on its removal, with the probable collapse

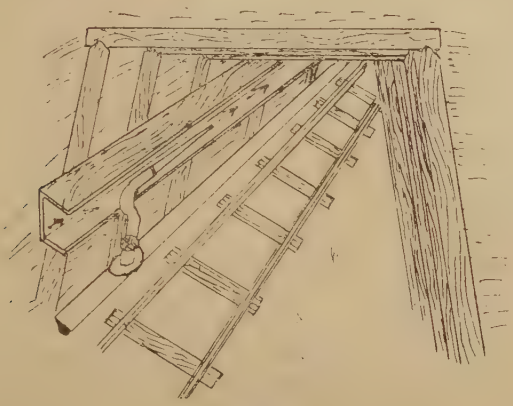


Fig. 69.

of the upcast shaft. A special shaft is usually needed, because temperatures of over 300° F. are obtained in some upcasts, but where the shaft is used for winding men the body temperature, say about 100° F., is the maximum permissible. The best

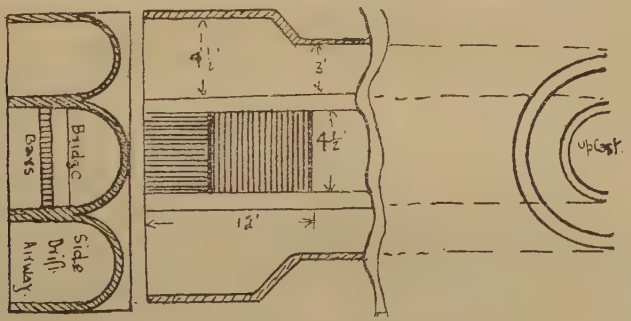


Fig. 70.

effects are obtained by making the deepest shaft the upcast, and in some cases having a surface chimney. Furnaces may be classified as follows :—

(i) Open furnaces.
(ii) Side drift furnaces, fig. 70.
(iii) Dumb drift furnaces, fig. 71.
(iv) Any combination of the above.

(i) **The Open Furnace** is simply an open grate, the whole of the return air passing over and through the fire. Its limitations are obvious.

(ii) **The Side Drift Furnace** consists of side or roof chambers separated from the furnace by the thickness of the brickwork. The advantage here is that the strata are not unduly heated and

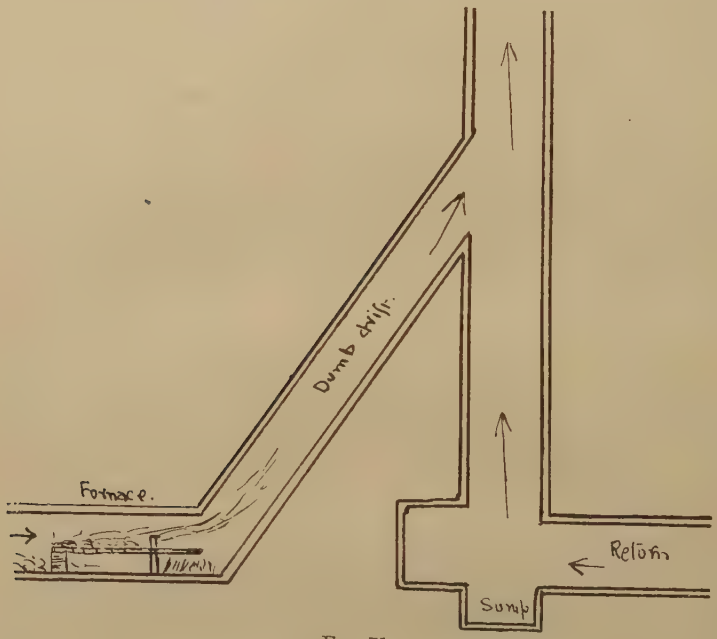


FIG. 71.

the current can be much better regulated, but it is expensive to construct.

(iii) **The Dumb Drift Furnace** method of ventilation consists of a furnace fed by a regulated split of air from the main intake so that only pure or fresh air passes over the fire. The hot gases may pass up an inclined drift intersecting the upcast at a distance sufficient to allow them to cool to within their ignition temperatures, or they may pass right up the upcast and the return air up the dumb drift. This method minimises the possibility of an ignition or explosion, and is enforced by the C.M.A., 1911 (sec. 31), *where furnaces exist*, except for mines where inflammable gas is unknown.

(iv) **Combination Furnaces** may be any combination of the above methods acting jointly, usually (i) and (ii), but surface furnaces are sometimes met with having compartmented shafts and a chimney. Duplicate furnaces, *i.e.* furnaces in parallel, are also adopted.

Furnaces act on the principle of (58), (60), etc., and use up to 500 lbs. of coal per hour or up to as much as $\frac{1}{2}$ lb. per 1000 cub. ft. of air passing per minute.

The average English coal has a calorific value of 12,500 B.Th.U. per lb., *i.e.* one pound will, by its combustion, raise the temperature of 12,500 lbs. of water through 1° F. The specific heat of air being about 0·24, it will be seen that a pound of coal will raise the temperature of $\frac{1}{0{\cdot}24}$ or about four times as much air as it will water through 1° F., *i.e.* about $4 \times 12{,}500 = 50{,}000$ lbs. of air or about $\frac{50{,}000}{0{\cdot}0766} = 650{,}000$ cub. ft. of air of ordinary density. A good coal capable of 14,000 B.Th.U. per lb. would increase this figure to about 780,000 cub. ft. in the same way. This is approximately the maximum. The low specific heat of air is, as the above shows, a providential thing for mines.

We can now state that for a furnace whose thermal efficiency is m, to raise the temperature of Q cub. ft. of air per minute from the absolute temperature T_1 of the mine to T_2 in the upcast, it will require to produce

$$\frac{cdQ(T_2 - T_1)}{m} \text{ units of heat} \quad . \quad . \quad . \quad (133)$$

c and d are the specific heat and density of the air respectively. Here m, the thermal efficiency, may be taken as $\frac{2}{3}$; it depends on the heat losses by radiation and conduction through the strata, which, in turn, depend on the natural conditions, *i.e.* presence of water, thermal conductivity of the strata, humidity, etc.

Allowing our previous figure of 12,500 B.Th.U. per lb. for the fuel the amount required will be—

$$\frac{cdQ(T_2 - T_1)60}{12500m} \text{ lbs. per hour} \quad . \quad . \quad . \quad (134)$$

E.g. (i) A furnace system produces a measured quantity of 43,620 cub. ft. of air per minute, and in so doing raises the air temperature by 80° F. What is the amount of coal required ?

$$L = \frac{\cdot 24 \times \cdot 0766 \times 43620 \times 80 \times 60 \times 3}{12500 \times 2}.$$

L = (462 lbs. of coal per hour, or 0·17 lb. per 1000 cub. ft. per minute).

Knowing the quantity of coal used, its calorific value, and also that 1 B.Th.U. = 778 ft.-lbs. of work, we may obtain the horse-power of the furnace—

$$\text{h.p.} = \frac{L \times 778 \times \text{B.Th.U.}}{33000 \times 60} \quad . \quad . \quad . \quad (134)$$

Having found the quantity of coal consumed per hour by the furnace, and making the general assumption that a furnace requires about half as much coal per square foot per hour as a Lancashire boiler, say 12 lbs., the dimensions of the furnace can be deduced.

The motive column is got by the method of (60) when the weight d of a cubic foot of air in the downcast is given; the ventilating pressure will be Md lbs. per sq. ft.

E.g. (ii) In a mine 800 ft. deep where 75,000 cub. ft. of air passes per minute, the temperature of the furnace upcast is 181° F. and the downcast 61° F. If the downcast air weighs 0·0766 lb. per cub. ft., find the h.p. of the furnace.

$$M = \frac{800(181 - 61)}{459 + 181} = 150 \text{ ft.}$$

$$p = Md = 150 \times 0{\cdot}0766 = 11{\cdot}5 \text{ lbs. per sq. ft.}$$

by (22) $$\text{h.p.} = \frac{Qp}{33000} = \frac{75000 \times 11{\cdot}5}{33000} = 26{\cdot}14 \text{ h.p.}$$

The efficiency, *i.e.* the ratio $\frac{\text{Useful work}}{\text{Work expended}}$, will be found thus—

$$= \frac{\frac{Qp}{33000}}{\frac{L \times 778 \times \text{B.Th.U.}}{33000 \times 60}} = \frac{60\ Qp}{L \times 778 \times \text{B.Th.U.}}.$$

In making the furnace calculations we assumed a heat efficiency of 66·6 per cent. This in many cases may be considered very generous, since a Lancashire boiler, which can be regarded as much more efficient than a furnace, has a heat loss of about

50 per cent.* Modern fans are usually over five times as efficient as a furnace and deal with quantities up to half a million cubic feet per minute. These facts, apart from any considerations of convenience and safety, place the furnace, with its maximum capacity of about 300,000 cub. ft. per minute, totally out of competition. Furthermore, the introduction of furnaces in newly opened mines is forbidden by the C.M.A., 1911 (sec. 31 (4)),

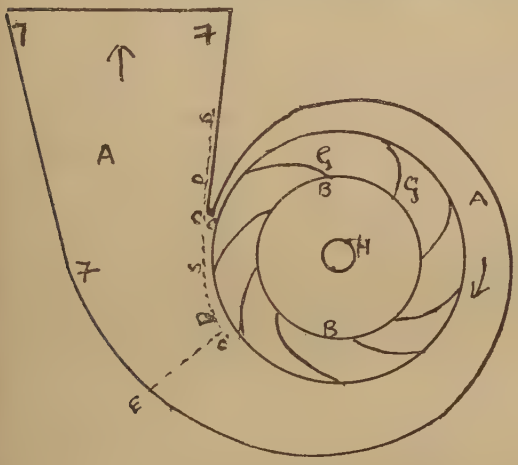


Fig. 72.

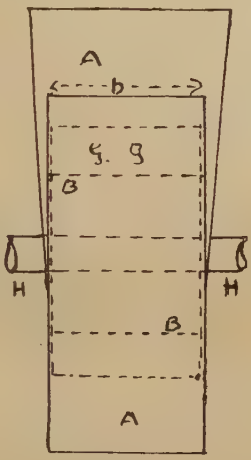


Fig. 73.

except in the case of a *small* mine with an upcast containing no inflammable gas.

(*d*) **Mechanical Ventilators.**—Mechanical ventilators are of many and varied kinds, and we will deal only with the class known as centrifugal ventilators or fans. These may be broadly classified as forcing or exhausting types dependent on whether they force the air *into* the mine from the top of the downcast or extract it *from* the mine when they are situated near the top of the upcast. The exhausting type is commonest in mines and is almost the only type used in Great Britain, the forcing type being adopted for cupola, forge, furnace, and propeller work in the metallurgical arts, etc.

The Salient Features.—Figs. 72 and 73 show diagrammatic sketches of a centrifugal ventilator. A A is the casing, which may be a volute, logarithmic spiral of equal width, or Archimedean

* D. Brownlie, *South Wales Inst. Eng. Proceedings*, xxxvii. 5, November 1921.

spiral of same width, the latter forms being common owing to their ease in manufacture. Casings may be made in brickwork, sheet steel, or concrete. B B is the inlet, which may vary from 30 per cent. up to 95 per cent. of the fan wheel side area. The casing sides are often called the cheeks or ears and are usually parallel, but in some types are bulged about the shaft axis to reduce axial eddies. C is the cut-off or beak with a clearance C C, which varies from 1 in. to 5 in. Small clearances develop surging or pulsations, and also create the liability to stripping when fans are quick-running and long bourne. Short spans between bearings eliminate this whip stress. D S, a sliding shutter, is often used for varying the cut-off ordinate C′ E. This ordinate C′ E must be at least—

$$C'E = \frac{\text{Quantity}}{\text{Area passed over by fan width}} \quad . \quad . \quad (135)$$

$$= \frac{Q}{\pi.n.d.b.}$$

Where Q = cub. ft. per minute.
n = r.p.m.
d = diameter, ft.
b = width, ft.

In the older types the width b is usually less than $0{\cdot}8d$, but in the modern fans it varies up to 1·8 times the *total* inlet diameter. F F is the expanding chimney, which may expand from 1° to 12° from the vertical on all faces, according to type, but in the Guibal type the flare is on the off face only and of about 7° to 8°, the near face being vertical and tangential to the casing at the cut-off. The height of the chimney should be such that the flare makes the area at the top at least four times that at the cut-off. The blades G G may vary in number from 6 up to 130, and their depth from a few inches at the rim down to full radius. H H the boss is specially shaped to guide the air to the vanes, and may be cast in halves with the spider, shrunk on complete, or built up. All fans should be duplicated and have duplicate engines, and may be driven by (i) clutched-in rope; (ii) belt with fast and loose pulleys; (iii) direct or friction clutches; (iv) a spare big-end drive on the opposite side; or (v) centrifugal clutches (Broad-bent). The fan-engine should be connected to a condenser if steam driven and run alternately at periods depending on the duty.

The Principle of the Centrifugal Ventilator can easily be comprehended if we imagine a particle of air being taken up by the

inner edge *a a* of the blades, *b b*, etc., of an exhausting fan (fig. 74), the fan under consideration being a number of fixed, straight-radial blades rotating with two attached parallel side-discs open at the inner and outer edges. The air having weight will be under the influence of centrifugal force and will therefore move outward to the periphery, where it will have the tangential velocity of the tips also. The centrifugal force causing the outward movement is proportional to the speed of the fan wheel. This in turn depends, by (21), on the square of the revolutionary speed and also on the radius, and therefore the diameter of the fan. The air on being ejected by this force is slightly compressed against the preceding body of air discharged, and in this way its density is increased and so helps to move the air in front of it. Thus there is in front of the vanes a region of higher pressure than within the fan. Modern fan chambers are so designed that this pressure shall increase as the air moves farther from the vane-tips into the chamber.

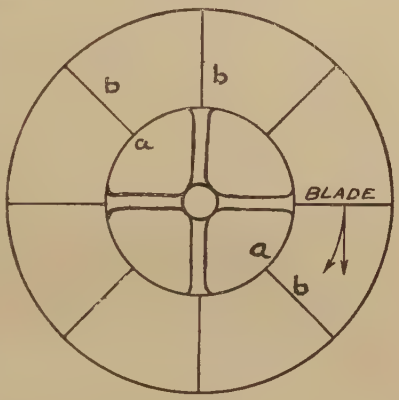


Fig. 74.

The Object of a Fan is to use the above velocity and pressure effects in the following manner: the greater the speed of the fan runner or wheel the greater the increase of pressure energy given to the air, consequently the greater tendency is there for a vacuum to be created at the centre of rotation. Since the mine is a closed system this region at the fan centre will possess the lowest pressure, the result being that the atmospheric pressure of the mine air will dominate that of the partial vacuum in the fan, the difference depending on the forementioned factors. The mine air, in an attempt to create natural balance, will rush in to destroy this partial vacuum and so be removed by the blades. This motion is transmitted throughout the mine—since it is all one "closed" system—and so a current is set up. The function of the fan is, then, *to set up a region of low pressure.* Since this pressure difference depends on the mechanical energy

expended and the resistance of the mine, such a fan to give the best results must—

(i) Possess very little inherent resistance.

(ii) Convert as much K.E. of the air into P.E. as it possibly can.

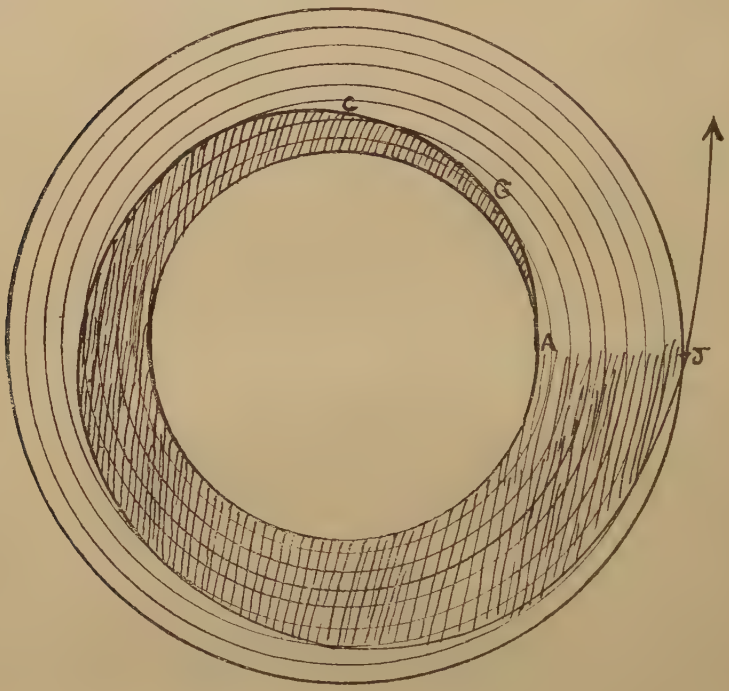


Fig. 75.

(iii) Spend as little energy as possible on work other than the creation of a partial vacuum.

(iv) Be mechanically efficient.

(v) Be situated at the end or low-pressure part of the ventilating circuit.

The Enclosure of the Fan.—The air arrives axially or endwise to feed a mine fan, and may do so from both sides, when it is known as a double-inlet fan. The axial motion of the air on being translated to a radial one means a right-angled turn to enable it to enter the blades. Now if the air was under the influence of an external force it would flow radially out through a

stationary fan, but as the only means available for creating the force are rotary ones, it will describe a path accordingly. The forces acting on the air, we have seen, are centrifugal and tangential, therefore it will be well to consider the air-track due to these outward and whirling forces as follows :—

Suppose the air has a radial velocity of 8 ft. per second and the wheel to run at 60 r.p.m., if we imagine concentric rings (fig. 75) to occupy the blading space, which we may take as 4 ft.

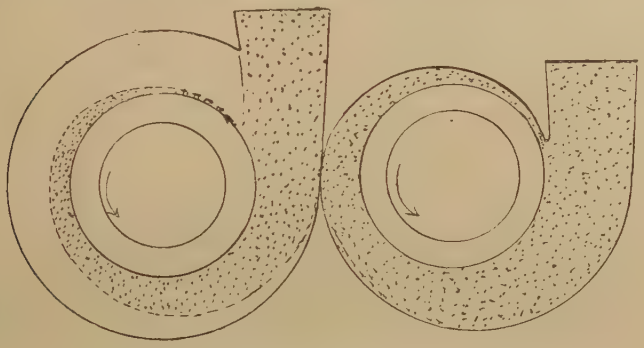


Fig. 76. Fig. 77.

deep, then a unit of air entering the blades at A will travel around to J in one revolution, in which time A returns to A, thus—

In $\frac{1}{8}$ second it will travel to G.
,, $\frac{1}{4}$,, ,, ,, C.
,, 1 ,, ,, ,, J.

The path joining these points is as shown in fig. 75.

It has been known since the researches of Guibal that a common shrouded disc or chamber with parallel sides, like the fan wheel under discussion, is very inefficient when discharging air all round its periphery directly into the atmosphere. This is due to the fact that the cross-sectional area of the fan increases outwards from the centre, but the air does not decrease in velocity (as might be expected), owing to its acquiring the peripheral velocity of the fan-tips. Re-entry of the air at the edges is the result, the outcome of which is a lower useful effect, to say nothing of the velocity energy squandered without being converted into pressure energy (to enhance the fan effect). An enclosed fan, *i.e.* one which revolves in an enclosed space or casing with one outlet minimises a great deal of the above loss. Fig. 75, showing

the path of a unit of air in the fan throughout a revolution, can also be applied to air ejected from the blades. It will be noted in fig. 76, which shows a fan placed concentrically in its casing, that the dotted space is the only part nominally filled with ejected air, the other space being wasted. This accounts for the placing of fan runners eccentrically in the casing, the cut-off being very close to the wheel, as in fig. 77. The whole space,

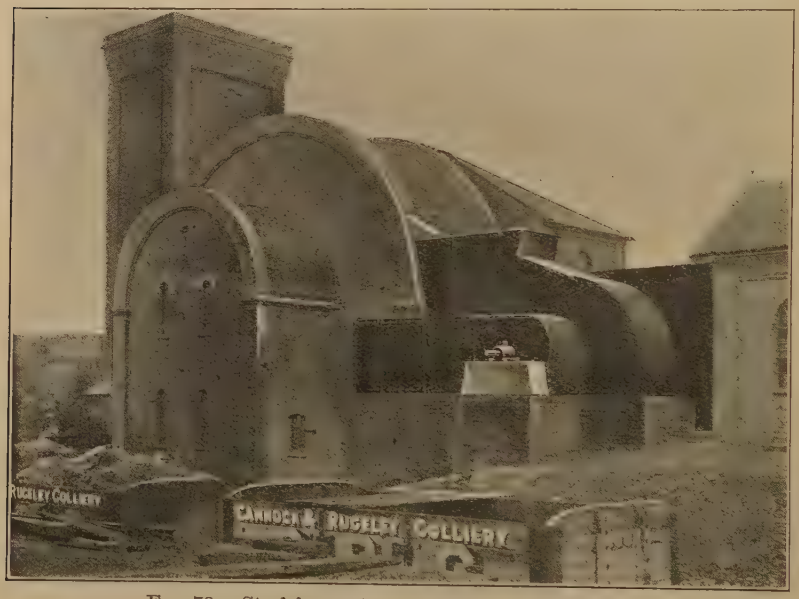


FIG. 78.—Steel fan casing (Bumstead and Chandler).

usually a volute, is by this means completely used and the Venturi effect is also fully gained.

The Evasée Chamber.—This is the name given to the expanding chimney (fig. 72), which starts from the cut-off of the fan C′, and continues the volute sometimes to a vertical tangent on one side. It is sometimes provided with a vertical shutter, cut to an inverted V-shape, which can be lowered or raised as desired, so restricting or increasing the cut-off area. The well-known Walker fans are fitted with this device, which, it is claimed, prevents vibration. The purpose of the evasée is to convert velocity energy into pressure energy. It thus forms a splendid example of the application of Bernouilli's theorem. An oper-

running fan, *i.e.* one which discharges direct into the atmosphere, is only half as efficient as an enclosed fan, because it can only set up half the pressure head, as will be shown later. (The Waddle fan, which is an efficient open-running fan, owes its success to its design. This fan declines in width from the centre outwards, so preventing peripheral re-entry of the air, and a bell mouthing at the tips gives an appreciable evasée effect.) The effect of the conversion of K.E. into P.E. at the outlet is to aid

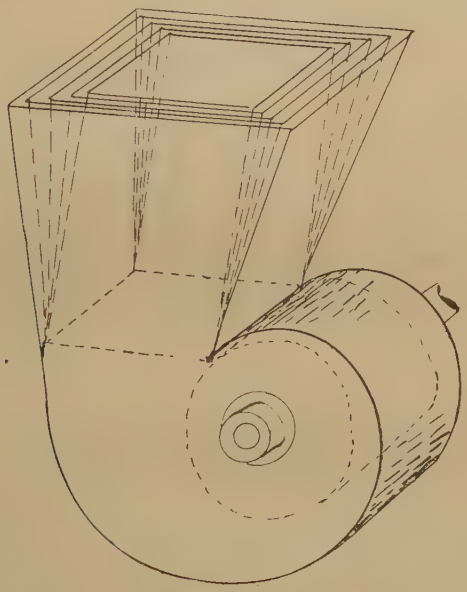


Fig. 79.

the fan, and an ideal ventilator would discharge the air into the atmosphere without velocity and so gain the full pressure effect.

Fan losses cannot be absolutely abolished, but good design will greatly reduce them. The energy loss at exit with which we are here most concerned depends almost entirely on the dimensions of the chimney, since it is proportional to the square of the volume passing.

E.g. if the outlet area be five times the cut-off area the velocity would be reduced one-fifth, and by (10), (96), etc., the energy remaining in the air to $(\frac{1}{5})^2 = \frac{1}{25}$th of what it was on entering the evasée. At the most the amount of the work done by the

fan existing as K.E. at the entrance to the evasée is 50 per cent.,* which is a high figure when we note the purpose of volutes. Therefore in the above case the K.E. carried away by the air would be $\frac{1}{2} \times \frac{1}{2\cdot5} = \frac{1}{5\cdot0} = 2$ per cent. at the most. If the evasée area were three times the cut-off area the untransformed K.E. would be 5·5 per cent.; if twice the area, 12·5 per cent. Note that the effectiveness of the evasée is proportional to the area. Again, if the evasée mouth and cut-off areas were the same—that is if we have no expanding chimney—the velocity, and

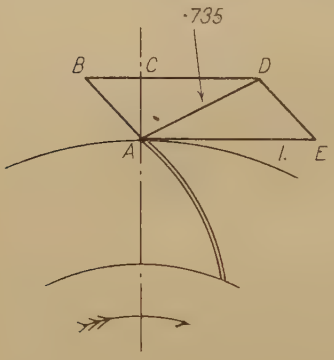


Fig. 80.

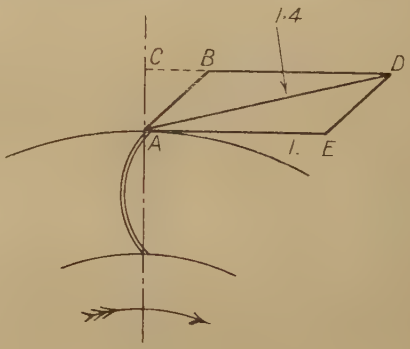


Fig. 81.

therefore the energy, wasted would be the same as the cut-off velocity and energy, *i.e.* 50 per cent. of the fan energy exerted. The failing of the open-running fan will now be manifest. Large evasées are built usually of brick, but sheet iron is commonly used, especially for small units. It is customary in this country to make a part of the casing volute of steel plates in order to facilitate the introduction and removal of the runner. The remainder of the casing and chimney is then practically an extension of the brickwork of the drifts.

Blading.—Fan blades may be straight or curved, in which case they may be curved backwards from the direction of rotation or forward, and may even be sinuous in shape, *i.e.* forward at the inner and backward at the outer part, or *vice versa*. Most of the older fans are slow-running, and have their blades curved backwards at the boss and few in number. They are also heavy and unwieldy, being up to 45 ft. in diameter. Modern fans are light, quick-running, shallow-bladed, of very large inlet, and have many

* Mr Joseph Parker, *Trans. Inst. M.E.*, lxii. 61.

blades. It will be seen by figs. 80 to 82, where the effect of blade curvature on the relative velocity and also quantity of the air leaving the fan is shown graphically, that the forward blades shoot the air out at a greater velocity than the peripheral speed of the fan. This increases the head or pressure effect of the fan. Properly designed fans must, in more or less degree, combat the following sources of loss common to all fans :—

(*a*) Loss of initial velocity due to the rectangular turn of the air on entering the blades and rebound against the central, or if single inlet the end, vertical disc.

(*b*) Eddying of the air at the periphery and in the casing.

(*c*) Friction of the air against the vanes and casing.

(*d*) Velocity loss of discharged air at evasée exit.

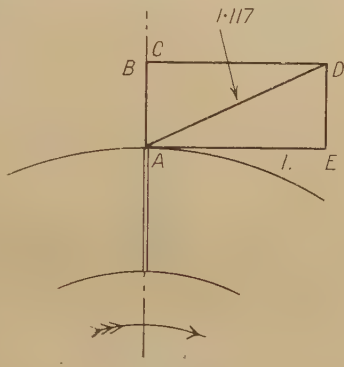


FIG 82.

Having dealt with (*d*), the remaining three, especially (*a*) and (*c*), depend chiefly on the blading, and since the whole of the work of a fan is done by the tips, much of the above loss can be reduced by cutting away the interior blading. By doing this the internal resistance is lowered to within 10 or 15 per cent., and fan inlets, formerly only 30 to 40 per cent. of the fan wheel, are now made practically as big as the wheel itself. The blades of the Sirocco fan are rarely more than 9 in. deep, or less than 64 in number ; in this way re-entry of air due to sparsity of blades is prevented. It will be seen from these departures that the losses (*a*), (*b*), and (*c*) can be greatly reduced, and that the scope of the modern fan is increased over that of an older fan giving the same pressure, so making it capable of working on a wider range of mine resistance and still retaining that pressure. The best fan will take up air at the inlet with the minimum of shock, conduct it smoothly over its vanes to the tips, where it will be ejected at the maximum velocity into the casing, to be finally reduced to the lowest possible velocity of discharge. Continental fan-makers appear to take greater pains in the design of fan blades, with a view to minimising the above losses, than British. The Rateau fan (figs. 170–173) shows these features well.

Apart from closed disc-bladed fans, unshrouded fans, *i.e.* those in which the sides are stationary and only the blades revolve, are used. The best-known British type is the Walker fan, fig. 83. This fan is entirely made of steel and is angle-iron braced to a built-up boss on a heavy shaft, and, like all centrally-bladed fans,

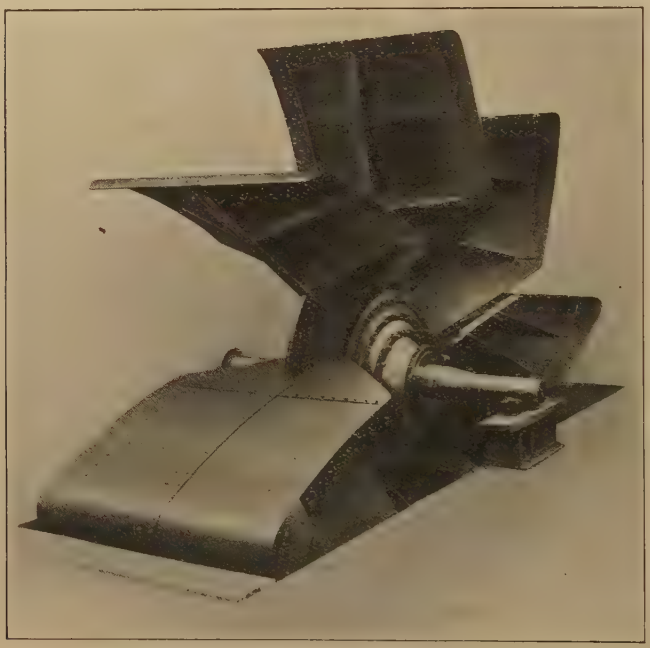


Fig. 83.—Runner of "Walker" fan.

attains a high peripheral velocity by increase of diameter. The blades are sheet steel and six to twelve in number. The cut-off has an antivibration shutter device.

Reversal.—The C.M.A., 1911 (sec. 31 (3)), requires that where a mechanical ventilator is employed a provision shall be made for the reversing of the air-current in the mine. This may be achieved in the following ways :—

1. **By installing an Independent Fan arranged to blow down the Upcast.**—This method is comparatively cheap, though one fan is lying idle at all times. In case of reversal the ordinary exhausting fan normally ventilating the mine would be slowed down, stopped, and isolated from the circuit by doors, the blowing

fan would then be started. It will be noted that no structural additions are needed for the downcast. An alternate method is to reverse the fan blading and casing if possible. It cannot be done with all types, but certain fans like the Turbon are built for easy structural reversal.

2. **By connecting both Shafts with a Drift,** the fan being placed intermediate between the two.—This method (fig. 84) works well if the shafts are fairly near to each other. A and A_1 are drift doors operated from the surface by levers a and a_1. B is

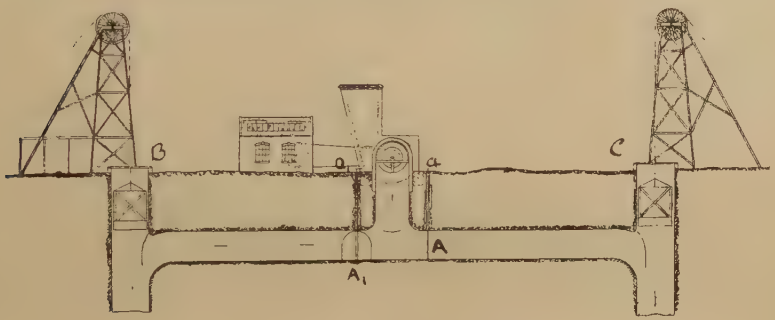


FIG. 84.—Reversal by drift.

the upcast and C the downcast. Normally air descends C, A being closed, ascends B, which is closed at the top, and passes through the open drift door A_1 up to the fan. On reversal C is covered over or closed, B is opened to the atmosphere, A_1 is closed, and A opened. The air now descends B, but cannot short-circuit through the drift since A_1 is closed, and so traverses the mine, passing up C, which is closed at the top, and through A up to the fan. Note that the direction of the air in the vertical fan shaft feeding the fan and of the fan itself is unchanged. The above idea as applied to a surface drift is shown in fig. 85, where B and B_1 are the ordinary intake and return. C is a closed road to the fan F, and D one to the atmosphere. The full arrows show the ordinary direction of the air. When the doors D_1 and C_1 are thrown over into the dotted position C is open to the fan, B is closed to the atmosphere, and B_1 closed to the fan, the current taking the course shown by the dotted arrows.

3. **By converting the Fan itself into an Exhauster or Blower as the case may be.**—This method is very popular in British mines and is applicable to any kind of fan, forcer, or suction, single or double inlet. Fig. 88 shows separate views of a Keith fan

arranged for reversal, and fig. 89 the same for a Sirocco fan. The procedure for reversal from exhausting to forcing is as follows: (1) the air-lock doors A, A_1 at the rear of the fan are opened to the atmosphere; (2) the front-drift doors B, B_1 are thrown over into the dotted position by means of levers external to the drift, so cutting off the fan from the mine inlets; (3) the hinged door C is swung up by levers into the dotted position shown in the cross-sectional elevations, thus closing the evasée. Reversal is now complete, the time taken being usually a minute

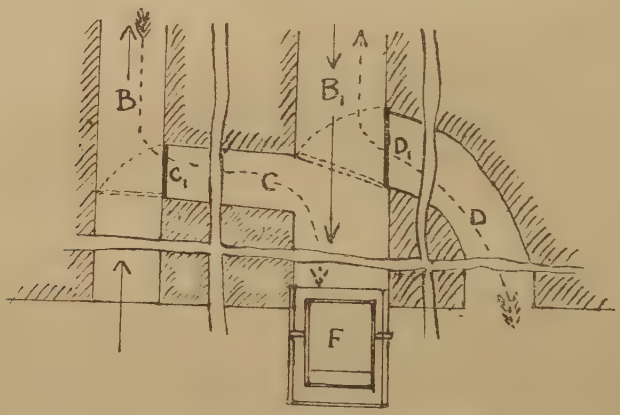


FIG. 85.—Reversing the air in a surface drift.

or so. The door C should be designed to give as much of the Guibal evasée effect as possible. Air now travels over the fork of the inlets into the fan drift and descends the upcast.

Figs. 86 and 87 show the method of reversing an open-running fan (Waddle). The door A closes the by-pass drift E when the fan is exhausting, but when it is required to blow fresh air down the upcast A is revolved into the position shown in both views, completely closing the main drift from the fan inlet. The door B is lifted admitting fresh air direct, and the flap-door D thrown over into the position shown in the elevation, thus closing the peripheral exit. Fresh air is now discharged along the by-pass drift from the fan wheel, and as the door A has now uncovered the by-pass the air passes into the main drift for the upcast. C, C' are air-lock doors for access to the fan, but if large enough can be used as supply doors instead of B.

Though reversal occupies very little time, the factors deciding

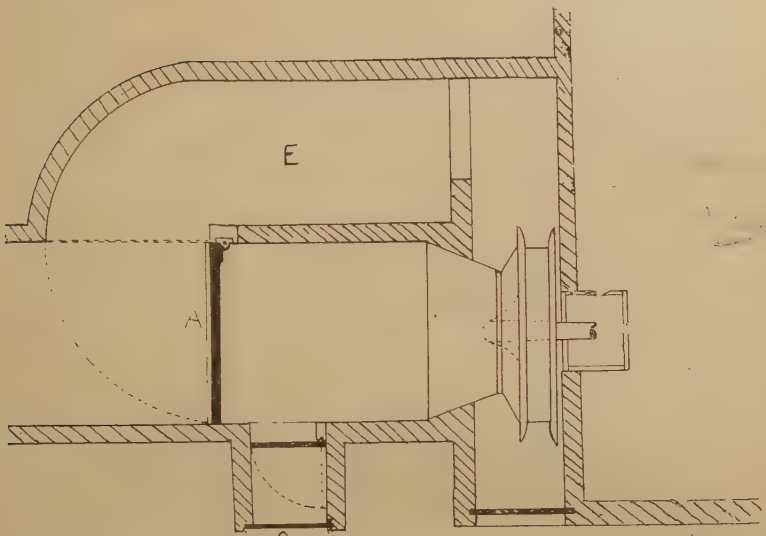


FIG. 86.

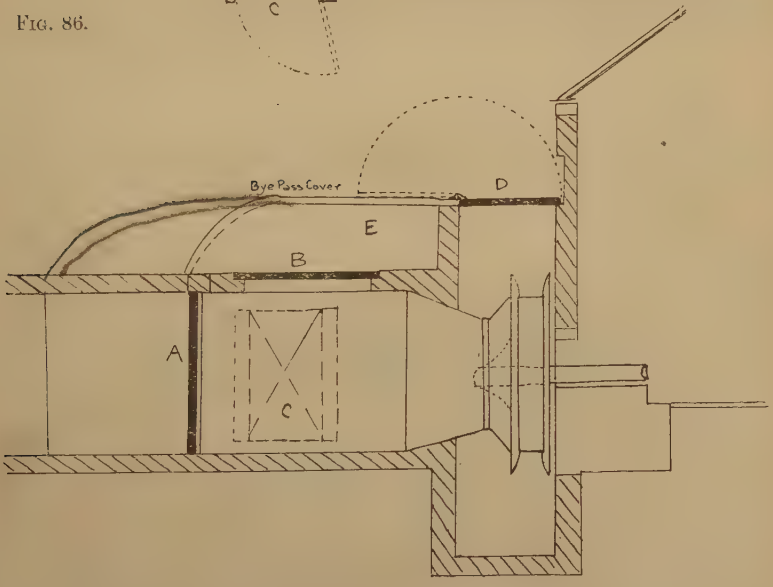


FIG. 87.—Reversal of open-running fan.

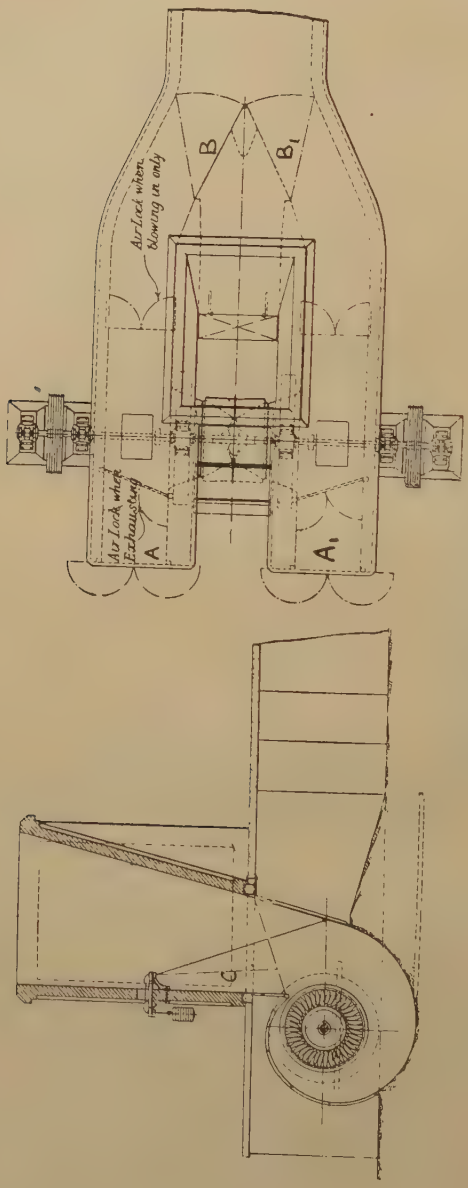


FIG. 88.—Reversal of a Keith fan.

its advisability need very careful deliberation. *The following are the occasions when reversal is usually resorted to :—*

(*a*) When it is desired to reach an underground fire without having to pass through the smoke.

(*b*) When it is desired to choke a fire with its smoke.

(*c*) When a fire occurs at the bottom of the downcast, in the downcast, in the fan drift, or in the main intake airway.

(*d*) Periodically, when the natural heat of the strata is high.

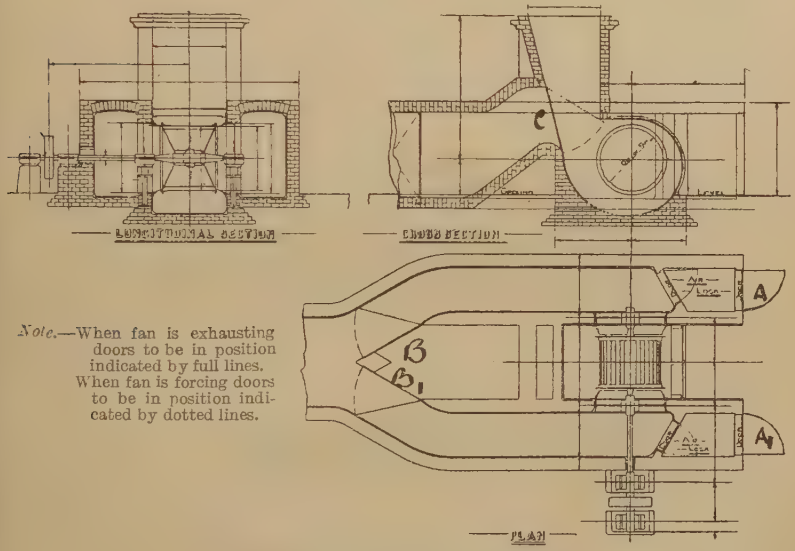

LONGITUDINAL SECTION
CROSS SECTION
PLAN
A
A1
B
B1
C
Note.—When fan is exhausting doors to be in position indicated by full lines. When fan is forcing doors to be in position indicated by dotted lines.

FIG. 89.*—Reversal of a Sirocco fan.

(*e*) In cold countries where the formation of ice in the intake impedes the ventilation; the hotter air of the reversal thaws the ice.

(*f*) When a fire breaks out on screens or top of downcast and is encroaching or likely to spread.

The equally important occasions and factors which demand consideration before reversing a fan are :—

(*a*) When a fire occurs in a mine which normally produces much firedamp. It would be extremely hazardous to pass the gas over a mine fire by reversal.

(*b*) When a mine fire burns incompletely producing much CO.

* By the courtesy of Messrs Davidson & Co., Ltd., Belfast.

Passing CO back over the fire and completing the action, producing CO_2, yields three times as much heat as the production of CO alone.

(*c*) Where the strata are friable and very oxidisable, normal reversal as an habitual practice causes falls by weakening the strata.

(*d*) Cold moist air has the same effect on certain kinds of returns as the above.

(*e*) Reversal as a dust remover from the leeward side of bars, etc., is not very effective.

(*f*) To reverse the fan means to effect an equivalent pressure in the reverse direction, *i.e.* if a fan forces at 5-in. pressure of water column and exhausts at 5 in. on reversal, this means 10 in. of water-column pressure *change* in the main airways, *i.e.* $10 \times \cdot 036 = \cdot 36$ lb. by barometer, *i.e.* 52 lbs. per sq. ft.—a very serious change for many mines. Gases now leave the strata more easily. The effect of this on a mine fire or even a normal mine should claim a first consideration.

(*g*) The danger of dust clouds by reversing is very great in some mines.

It will be manifest that the decision to reverse a fan depends in very small measure on whether mine doors, etc., will operate or tend to short-circuit, but rather upon vital issues which are supplied by each particular case.

The Position of a Ventilator in the Circuit.—In this country fans are, as required by law, always placed at the surface to avoid the risk of a ruined ventilating system in case of an explosion, but in metalliferous mines, in the U.S.A. and South Africa especially, they are frequently placed below ground. Boosters or auxiliary fans underground assisting the main unit are fairly common in Great Britain. *The advantages to be gained by placing a main fan below ground are—*

(i) Leakage through shaft top doors either by drawing in air in the exhausting method or by forcing some air out in the forcing system is done away with.

(ii) There are no shaft doors at the surface to impede winding operations.

(iii) The fan combines in some measure what advantages are to be got from both forcing and exhausting methods.

The disadvantages of placing a main fan below ground apart from cost of fan and engine chambers are—

(i) Danger of fire and wreck in case of an explosion.

(ii) Danger of isolation in case of a winding mishap.

If the fan is placed inbye, one of the greatest drawbacks (and an objection often raised against such a practice) is the liability of the air to short-circuit after it has left the fan. With a fan at the end of the circuit, one always has the knowledge that the whole of the air pulled by the fan is finally expelled from the mine. A fan placed in the circuit is not only an exhausting fan, but becomes a blowing or forcing fan also over the latter part of the circuit, and it becomes a very difficult matter indeed (especially in mines with large open inaccessible areas) to prevent the air taking the line of least resistance, the fan pulling the air back to it rather than pushing it to the surface.

The inability of a fan to discharge into the atmosphere without final air velocity destroys to a certain extent its ability to create depression, or, in other words, reduces its efficiency. This to a certain degree can be remedied with a fan on the surface by exhausting into an expanding or evasée chimney, as we have already seen. What really happens with a fan underground is that the air is delivered at a great speed against an excessive resistance, the tendency being to compress the column, throwing an excessive back pressure on the fan, materially reducing its efficiency.

The practice with a fan at the end of the circuit is to have one leg of the water gauge open to the inlet side (fan chamber), and the other to the outlet (atmosphere). A water gauge so placed on an underground fan, one leg open to the inlet and the other to the outlet, does not give the effective depression or W.G. producing ventilation, but one which is excessive. With a fan, say 4000 ft. from the surface, the reading of a W.G. so placed may be as much as twice the correct or actual W.G. producing ventilation.* The position is the same as with an underground furnace producing ventilation, and the correct position of the water gauge is to have one leg open to the fan chamber as before, and the other connected to the intake side of the dividing doors at that point.

Forcing *versus* Exhausting Fans.—The old methods of forcing by isolating units of air and moving them into the mine by means of pistons or rotating doors will not be discussed here, since they are passing from the industry and are so very limited in capacity. The ordinary mine fan can, we have seen, be arranged for blowing or exhausting without affecting the general design, power, or quantity, except that the casing may have to be altered. Blowing

* E. C. Ransom, *S.A. Min. and Eng. Journal*, 21/5/21.

fans have the advantage of maintaining a higher mine pressure than that of the atmosphere, so preventing the accumulation of gases to a greater extent than an exhausting fan, which creates a lower pressure than that of the atmosphere. The increased density set up by the former assists the cooling and drying power of the air. In a very dusty mine ventilated by an exhausting fan, where the haulage is on the main intake, the air is liable to get very foul before reaching the face, which, if a forcing fan were employed, would not be so serious, as the same road would then be the return. On the other hand, exhausting fans are most used, because when the fan stops the mine air pressure rises, whilst in the case of the blowing fan it would fall to that of the atmosphere and so be favourable to the emission of gases from the strata. Again, in the case of a mine with a gassy return, the best system is the exhausting method. In this system the fan is placed over the shallower shaft, whilst in the forcing method it is placed on top of the deeper shaft, which is usually the winding shaft. Choice of type will depend on local circumstances and expediency, but in this country the exhausting fan is almost entirely used, due largely, among other things, to tradition. Fig. 90 * shows the design of forcing fan wheels with and without the diffuser. Forcing fans are very successful for engineering work where they are made to impart pressures from 0·5 in. up to 56 in. of water gauge, when the deep-blading problem becomes difficult. Their efficiencies are usually from 50–65 per cent. Exhausting fans seldom give over 10 in. W.G., and usually about $2\frac{1}{2}$ in. It is considered good practice to keep mine-

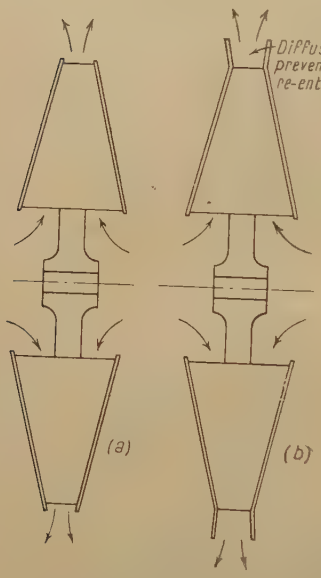


Fig. 90.—Forcing fan wheel.

* By Messrs Standard & Oswald Stott, Engineers, Leicester.

ventilating pressures down, because, among other things, leakage (through goafs, etc.) increases *directly* with the pressure.

Auxiliary Fans Underground.—Occasions sometimes arise in mines when sec. 42, C.M.A., 1911, with regard to the maintenance

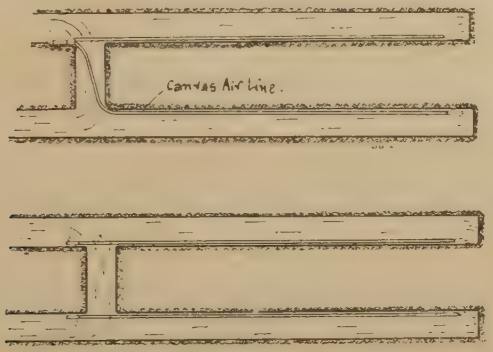


FIG. 91.

of two main intake airways becomes inoperative, when exemption is provided. Among such occasions may be mentioned narrow exploratory drives towards old workings suspected of holding dangerous bodies of water or gas. In such cases auxiliary venti-

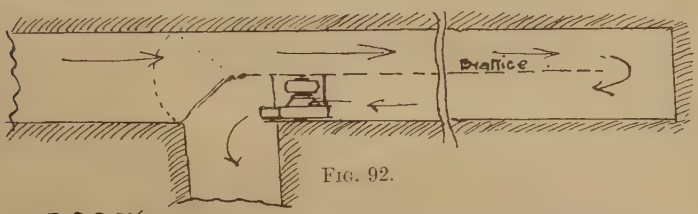


FIG. 92.

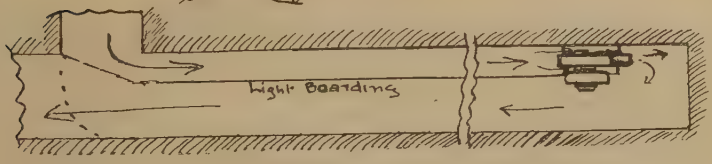


FIG. 93.

lation is often employed as an advance over the old method of plain coursing by piping, brattice sheeting, or light partitioning of an intake and return in the same roadway, especially when the distance becomes great. Other and frequent occasions are when the air current is sluggish, the auxiliary means being

employed for a certain district or the whole mine as an aid but not as a substitute (see sec. 31*b*, C.M.A., 1911).

Figs. 91–93 show the position of forcing and blowing boosters. Fig. 94 shows the mode of construction of wooden ventilating troughs.

(i) **Auxiliary Fans, Axial Propulsion Type.**—The theoretically correct position for an auxiliary fan is in that part of the mine where equilibrium of pressure exists, *i.e.* no water-gauge difference, but whether this is the most suitable place depends upon the

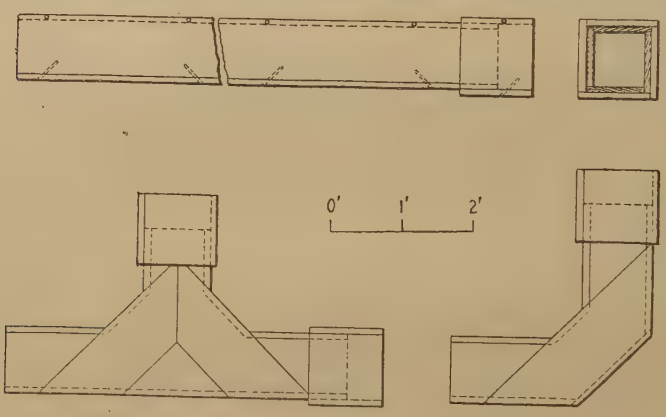


FIG. 94.

geography of the mine and other local factors. These fans may be of the ordinary peripheral discharge type for exhausting or forcing, or they may be axial forcers on the screw blower principle. Boosters of this latter type working in conduits are sometimes adopted, but their great drawback is that they provide low pressures, needing several in series for a long conduit, and they also have somewhat limited volumes. They can be arranged to compress and exhaust simultaneously, and in every case their delivery must be fitted with a conical branch of definite shape wherein the kinetic energy of the air is converted into static pressure. Axial propulsion blowers possess the following advantages for auxiliary ventilation purposes :—

(1) Compact axial construction permitting of placing the unit in a straight line with, and at one end of, the pipe line, or, alternatively, inserting it in the line.

(2) High speed, resulting in a small driving motor.

When using galvanised sheet piping with inserted joints, air

leakage becomes a very serious source of loss in a long line. Considerable reduction in leakage loss can be effected by arranging the fan for pressure working up to half this length and then carrying the pipe line the remainder of the distance as a suction. The latest blower of this class, the "Schlotter," by Messrs Siemens-Schuckert, consists of about five blades working in a stationary guide drum having eight or ten blades, the air being propelled axially (see fig. 95). The pressure is $7\frac{1}{2}$-in. W.G., and the efficiency

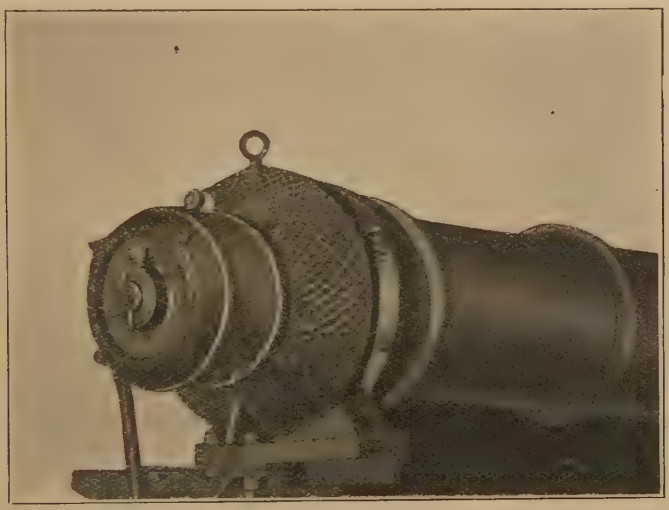
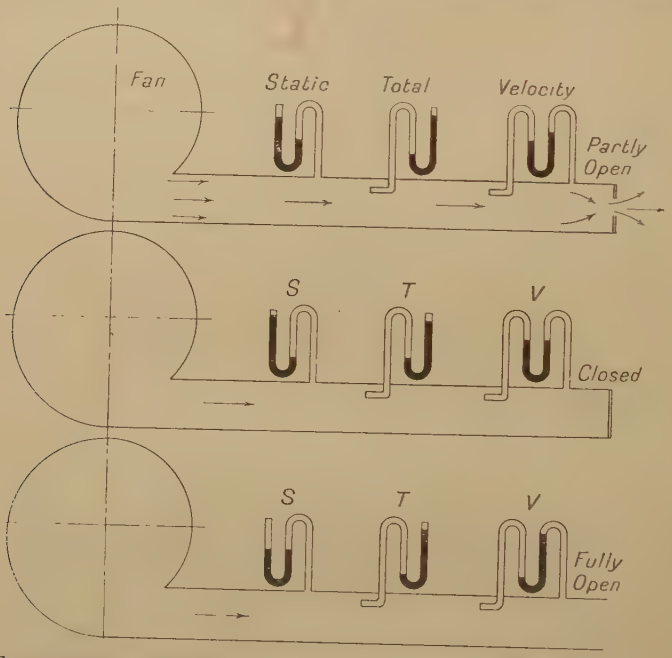


FIG. 95.—Schlotter blower.

high for the type, 80 per cent., which is due to the cross lie of the moving and fixed blades, the latter increasing in curvature in the direction of rotation, so giving a radial subdivision to the air, which is here accelerated and leaves without shock. On starting the fan a thin conical jet of air will leave the fan axially, the apex of which contracts nearer to the guides as the speed rises, its base increasing radially, so creating pressure and consuming a uniform amount of energy which remains constant over a wide load range. Thus, where the counter pressure cannot be exactly predetermined, or where the length of the delivery conduit is varying as the drive progresses, an adequate volume of air can be continuously relied upon without any danger of overloading or burning out the driving motor.

The blower is combined into a single light unit with a motor

running at 3000 r.p.m. The motor is usually a three-phase squirrel-cage unit, totally enclosed, and thus effectively protected



FIGS. 96, 97, and 98.—Effect of varying orifices on W.G. (forcing fan).

from the deposits so common at the inlet of a blower. The fan inlet is protected by coarse meshed screening.

The smallest size blower of this type will deliver 400 cf/m

FIG. 99.—Jeffrey booster.

through a 10-in. pipe 750 ft. long with 0·5 h.p. electrical input, and the largest size will give 3200 cf/m through 3600 ft. of 24-in. pipe, or twice this quantity through 850 ft. of the same pipe with an electrical input of 7·5 h.p.

In cases where the use of electricity is restricted the Turbon axial blower is often used. It consists of a similar fan in principle to the above, but derives its rotary motion from the impinging

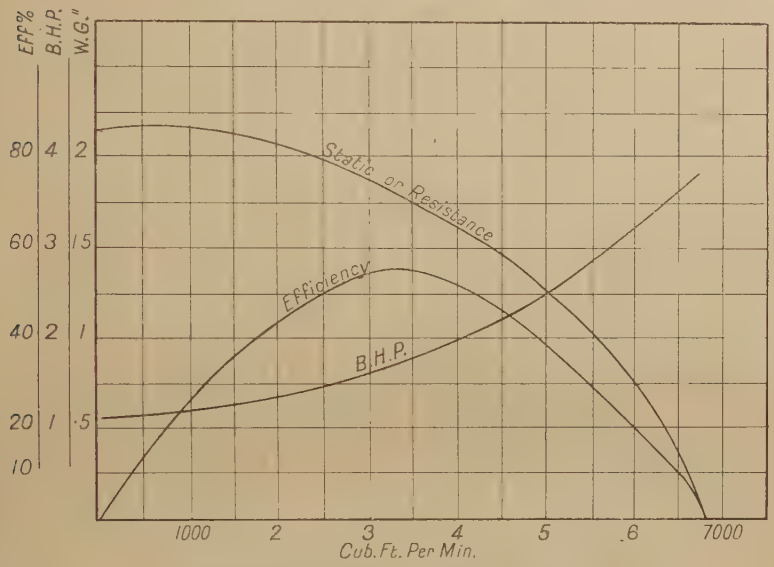


Fig. 100.—Characteristic curves for a forcing fan.

of a compressed air jet on the vanes, this air afterwards passing through to augment the supply.

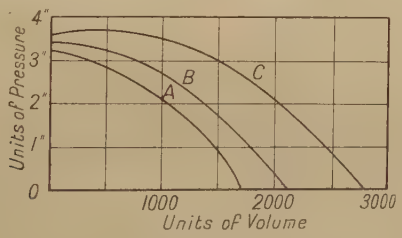
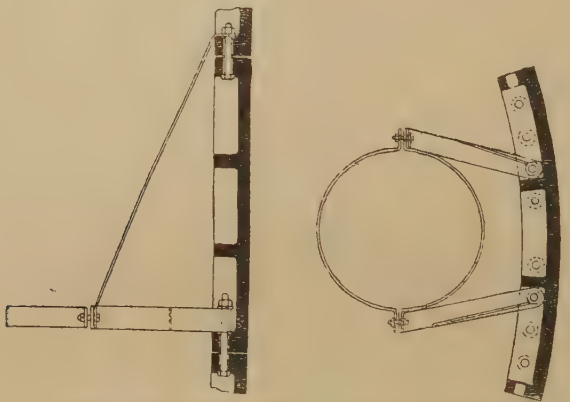


Fig. 101.

(ii) **Auxiliary Fans, Peripheral Discharge Type.**—Figs 96 to 98 show the pressure effect under different loads of a peripheral discharge auxiliary fan forcing into a metallic conduit. Figs. 100 and 101 show the characteristic curves for such fans.

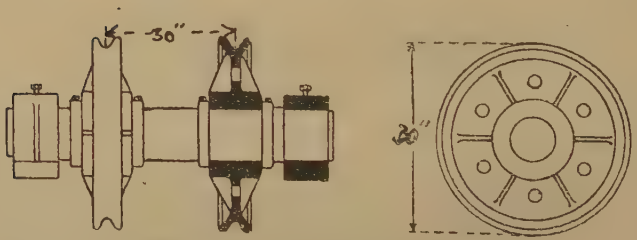
These fans are commonly used for auxiliary purposes in this

country owing to their ability to deal with larger volumes of air than the axial propulsion type. Fig. 99 shows a Jeffrey fan of this class arranged as an auxiliary blower. Fig. 112 shows a



FIGS. 102 and 103.—Attachments to tubbing.

FIG. 104. FIG. 105. FIG. 106.



FIGS. 107 and 108.—Rope pulleys.

Sirocco fan arranged for auxiliary work as a blower in the ventilation of a sinking shaft, and figs. 102–112 show shaft sinking ventilation accessories.

Small Fans for other Purposes.—It appears to be appropriate here to mention the chief places about mines where fans are used for other purposes than the ventilation of the mine.

Induced Draught for Boilers.—Figs. 113 and 114 show in plan and elevation two methods of installing peripheral discharge blowers in order to create a current and so put the attendant in

FIG. 109. — String of piping, with suspension hooks and cradle.

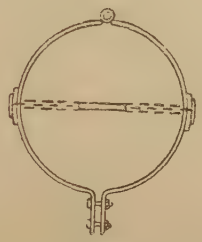


FIG. 110.—Sling for lifting air pipes out of shaft.

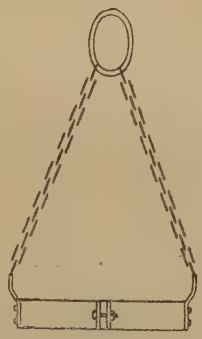


FIG. 111.

a position to regulate easily the rate of combustion at the boiler furnaces. The figures, which are supplied by Messrs Matthews & Yates, of Manchester, being sufficiently detailed, no further explanation will be needed.

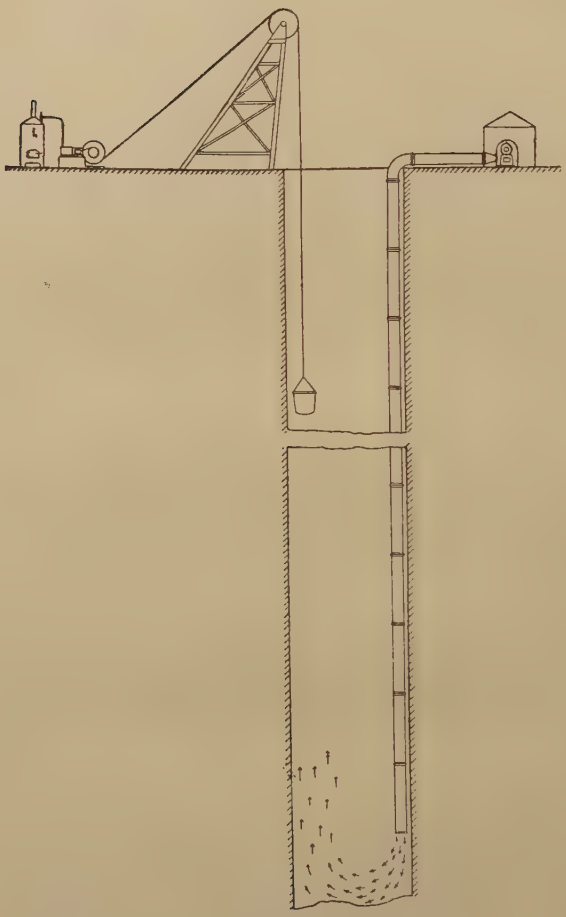


FIG. 112.—Auxiliary ventilation for sinking shaft.

FIGS. 102 to 112.—Showing ventilating accessories in sinking shafts.

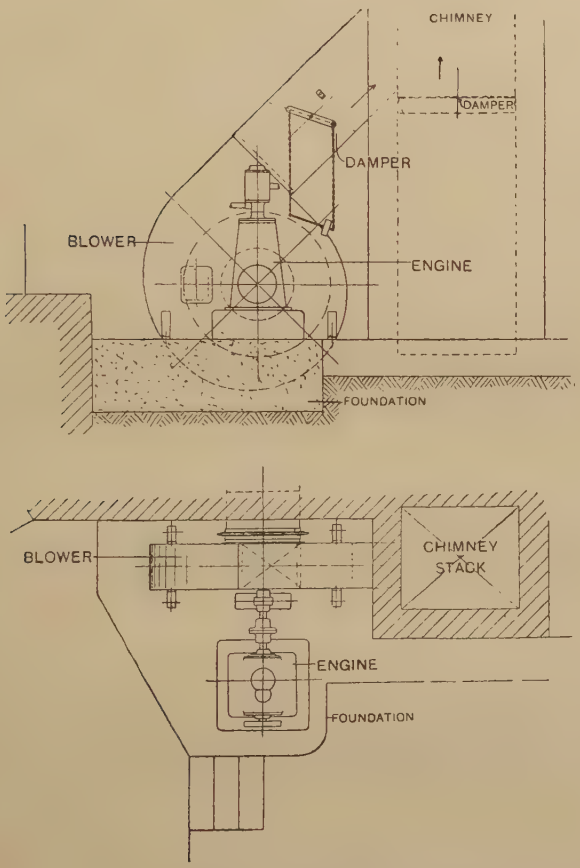


FIG. 113.—Induced draught fan for boilers.

Dust Collectors for Screens.—At exposed collieries the screen dust nuisance occasionally becomes a menace to the ventilation, especially on the downcasts of old mines with near screens,

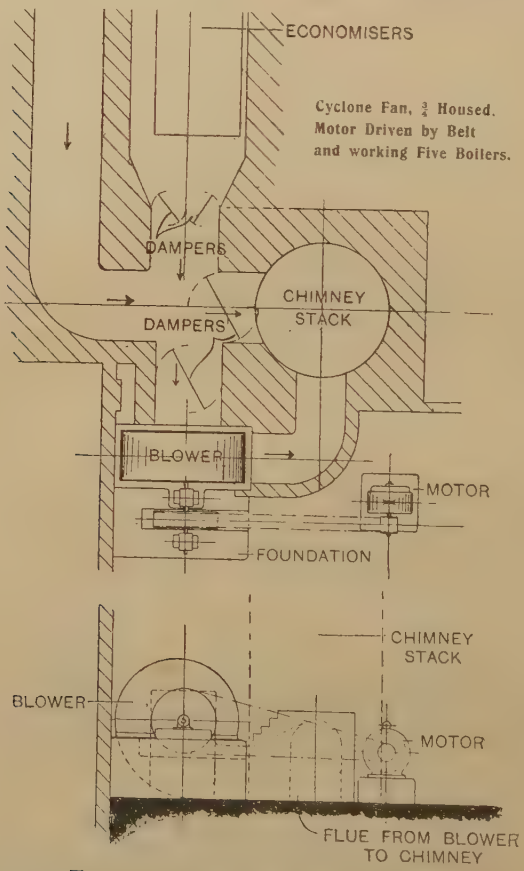


FIG. 114.—Induced draught for boilers.

and also on bonneted or otherwise partially open upcast tops in very bad weather.

Fig. 117 illustrates the principal features of a dust collector.

The dust-laden air is drawn through the pipe A from the point of generation B into the conduit C by means of the exhausting fan D, the inlet of which is connected to C. After passing through D the dusty atmospheres are led to the inlet of a pressure fan E,

which forces it into the main F for the separators or for use in other places according to the nature of the dust. The cyclone collector has no separate pressure fan, the exhauster being also the forcer over the last part of its circuit.

Figs. 115–116 show in sectional and full elevations the cyclone

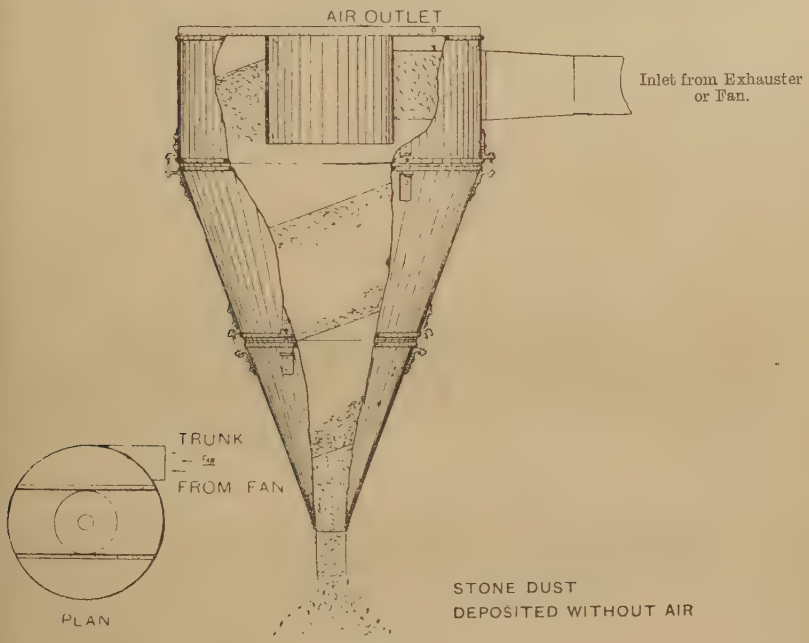


Fig. 115.

separator of Messrs Matthews & Yates, of Manchester. Its purpose is to deposit effectively the collected dust without a cloud. The principle of the spiral separator will be immediately seen from the sketch and therefore needs no further elaboration.

Fig. 118 shows an application of the cyclone dust-collecting system to the screens of an exposed colliery in South Wales. Specially designed sheet-steel covers and hoods are fixed over the tipplers and screens, practically boxing them in, and piped to the main suction trunking; another connection is taken from the boot of an elevator. The main trunking is carried beneath the tippler platform to a heavily constructed cyclone exhauster in a lock-up room beneath the screen level. It is belt-driven

from an electric motor. The dust is separated by a series of separators of the type dealt with above, and the leaving air

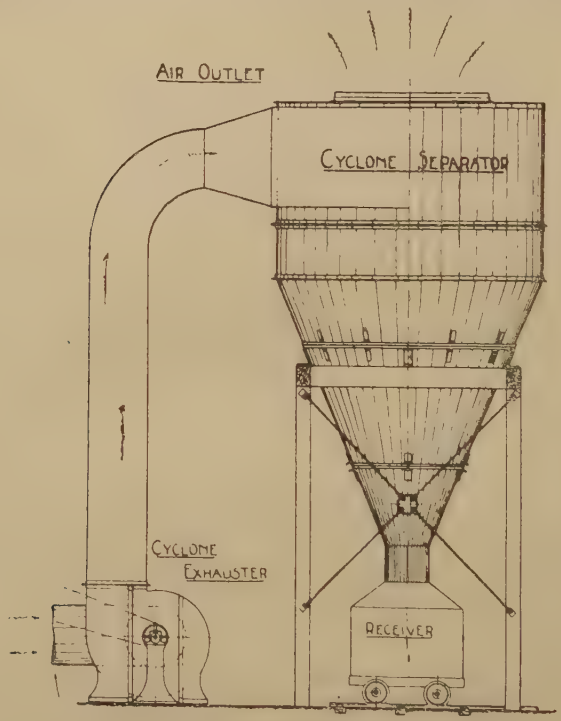


FIG. 116.

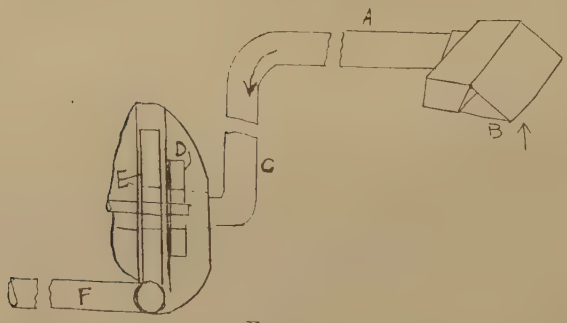


FIG. 117.

discharged high above the roof. 7 to 8 cwts. of dust are collected daily; this is used for the boilers.

Air Filters.—These are used to clean the air of solid impurities

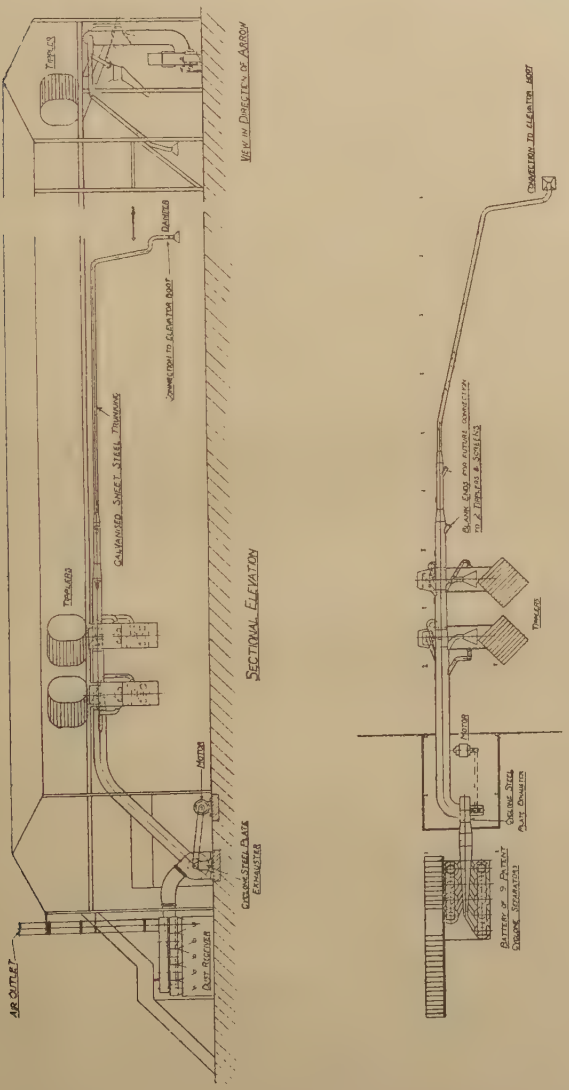


Fig. 118.—Dust collectors on screens, etc.

previous to using it in particular cases, as in the ventilation of turbo-alternators and other special purposes. The construction

is simple, as will be seen on referring to fig. 119 below. Two or more rotors, R, built up of discs of fine-mesh expanded metal, are mounted on a shaft working through worm gear, G. The rotors dip into water for a depth of a few inches and revolve at a slow speed. Eliminators, E, at the outlet of the machine, prevent any excess moisture leaving the machine. The water lost by evaporation is very little and is made up by a ball valve and constant-level arrangement.

As the rotors pass through the water they automatically pick up sufficient to wet thoroughly and cleanse the whole surface.

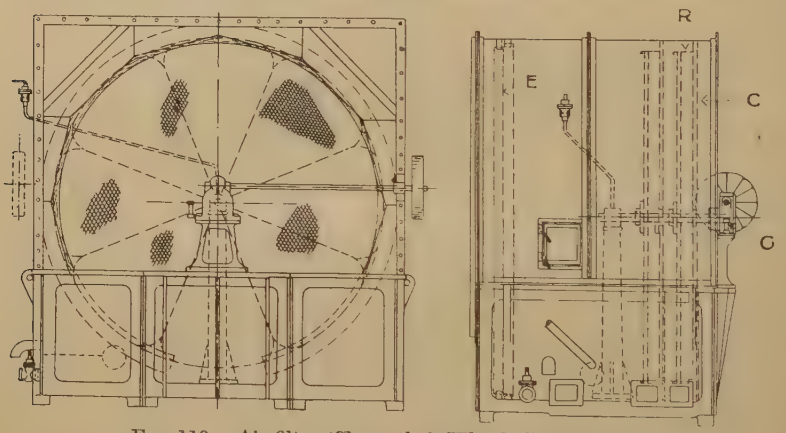


Fig. 119.—Air filter (Cleworth & Wheal, Castleton).

The mesh of each disc of expanded metal is set in such a position as to give a baffling effect to the air, and so the course of the air current through the revolving wet mesh is tortuous. The air thus undergoes a scrubbing action, and as the depth of water is only a few inches the resistance in water gauge is small.

Ventilation Efficiencies.—By the efficiency of a machine is meant the ratio of the output to the input. It is usually expressed as a fraction of unity or as a percentage, *i.e.* it is the numerical value of the effectiveness.

The chief efficiencies dealt with in mine ventilation are—

(*a*) The manometrical efficiency, *i.e.* the true mechanical efficiency.
(*b*) The overall or total efficiency, *i.e.* the useful effect.
(*c*) The drive efficiency.
(*d*) The air-system efficiency.
(*e*) The volumetric efficiency.

(*a*) **The Manometrical Efficiency of a Fan is the Ratio of the Observed to the Theoretical W.G.**—We have seen by (35) that when a weight, W, in lbs. (which in this case is air) is moved at a velocity of V ft. per second $\frac{WV^2}{g}$ ft.-lbs. of work is being done per second. Now confining the discussion to radially bladed fans, for reasons which we will discuss later, we see that each pound of air thrown out by the fan receives an additional $\frac{V^2}{g}$ ft.-lbs. of energy from the fan. Expressing this as a head H_1 by (96) we note that H_1 is the pressure produced by the fan and expressed as a head or column. If the air weighs d lbs. per cub. ft., then the pressure on the fan is $\frac{V^2d}{g}$ lb. per sq. ft., and converting this into inches of water gauge by (93) we get the theoretical water gauge as in (102),

$$\frac{12V^2d}{gd_1} = \frac{V^2d}{5{\cdot}2g}.$$

Calling the registered water gauge in the fan drift W.G., the manometrical efficiency

$$= \frac{W.G.}{\frac{12V^2d}{gd_1}} = \frac{W.G. \times g \times d_1}{12V^2d} = \frac{5{\cdot}2W.G.g}{V^2d} \quad . \quad . \quad (136)$$

Knowing the quantity of air, Q, in cubic feet of air per minute passing in the mine we can now obtain the *horse-power in the air* by (26) and (94), thus—

$$\frac{pQ}{33000} = \frac{5{\cdot}2 \times W.G. \times Q}{33000} \quad . \quad . \quad . \quad (137)$$

But by (36) the h.p. supplied to the air is $\frac{WV^2}{550g}$, so that the true mechanical efficiency

$$= \frac{5{\cdot}2 \times W.G. \times Q}{33000} \div \frac{WV^2}{550g} = \frac{5{\cdot}2W.G.Qg}{60WV^2} \quad . \quad . \quad (138)$$

Now $Qd = 60W$, which is fundamental, since Q is the quantity of air passed per minute, d its density, and W the weight passed per second. Thus by substituting in (138) the true mechanical efficiency can be stated as $\frac{5{\cdot}2W.G.g}{dV^2}$ . . . . . (139)

Note here that (136), (138), and (139) are similar. Hence the very important statement by Professor Briggs, which in effect is as follows: the manometrical efficiency of a fan is the same thing as the true mechanical efficiency derived by considering only the fan and its passages, *i.e.* no engine losses are taken into account. "A better criterion of the design of a fan does not exist, since it compares work output and work input without considering outside losses in the drive." *

(*b*) **The Total Efficiency,** which is the overall efficiency of the fan and the driving plant, has been in the past expounded and used somewhat carelessly. Many people consider this to be the mechanical efficiency of the fan, whilst really it includes losses in the drive, bearings, and other places, thus giving the *useful effect* of the ventilating system. It is the ratio between the horse-power put into the air and that supplied to the engine. The h.p. in the air is $\frac{5{\cdot}2\text{W.G.Q}}{33000}$, as already stated, and that given to the engine is got from $\frac{\text{PLAN}}{33000}$, where P is the mean effective pressure of steam in lbs. per sq. in., L the stroke in feet, A the cylinder area in sq. in., and N the number of effective strokes of the engine per minute, thus—

$$\frac{5{\cdot}2\text{W.G.Q}}{33000} \times \frac{33000}{\text{PLAN}} = \frac{5{\cdot}2\text{W.G.Q}}{\text{PLAN}} \quad . \quad . \quad (140)$$

It will be seen from the above that the manometrical efficiency of any type of fan we choose will be always greater than the total efficiency of the ventilating system, since the latter has to account for the frictional losses in the engine and drive in addition, whilst the former only deals with the fan and mine.

(*c*) **The Drive Efficiency** expresses the amount, or fraction, of the effort of the engine given out by the fan as useful work apart from that lost in friction in the bearings, belting, etc., and is

$$\frac{\text{Total efficiency of plant}}{\text{Manometrical efficiency}} = \frac{d\text{QV}^2}{\text{PLAN}g} \text{ or } \frac{60\text{WV}^2}{\text{PLAN}g} \quad . \quad (141)$$

(*d*) **The Air-System Efficiency.**—Murgue adopted an efficiency in terms of the fan and mine resistances thus by (111), etc.,

$$\frac{h}{\text{H}} = \frac{h}{h+h_0} = \frac{o^2}{o^2+a^2} \quad . \quad . \quad . \quad (142)$$

* *Colliery Guardian*, reprint from *Col. Guard. Pocket-Book*, 1910, and Prof. Briggs, *Trans. Inst. M.E.*, xliv. 251.

Mr Penman calls this the "aero-dynamic efficiency," and illustrates its usefulness in dealing with fans working in combination.* We give this efficiency here so that the student may derive the great advantage to be gained by proving its relation to the manometrical efficiency. $\left(\text{Note } o^2 \propto \frac{1}{h_0} \text{ and } a^2 \propto \frac{1}{h}.\right)$

(*c*) **The Volumetric Efficiency.**—This is the ratio of the actual volume discharged per revolution to the cubical contents of the fan. It is used chiefly on isolating forcing systems which may be reciprocating on the pump principle as in the Nixon ventilator, or rotary as in the Roots' blower or Fabry's ventilator. It varies up to 90 per cent. for the above type of ventilator. It is of more value to manufacturers than mining men, hence its comparative disuse. If the ventilator produces a quantity Q_1, from a theoretical displacement Q (*i.e.* volume of chamber multiplied by the number of times chamber is emptied per minute), Q_2 being the volume lost due to leakages, back pressure, etc.

$$Q = Q_1 + Q_2 \quad . \quad . \quad . \quad . \quad (143)$$

If a is the equivalent orifice of the mine and o that of the ventilator representing total leakage, loss, etc., as an area, then

$$\frac{Q_1}{Q_2} = \frac{a}{o}$$

$$\therefore Q_1 = \frac{Qa}{a \quad o} \quad . \quad . \quad . \quad . \quad (144)$$

and the volumetric efficiency is

$$\frac{Q_1}{Q} = \frac{a}{a+o} \quad . \quad . \quad . \quad . \quad (145)$$

The Volume Ratio, which is similar to the volumetric efficiency, is the ratio of the quantity of air delivered per minute by the fan to the product of the cubical capacity of the runner and the r.m.p. It varies from 50 per cent. to 300 per cent., dependent on the type of fan; with forward-bladed fans 200 per cent. is common. It is a convenient means of applying the above efficiency to centrifugal fans.

Fan Drives.—The fan engine is one of the most economical engines on the colliery, since its load is fairly constant and it is thereby well suited for condensation. It should be made so

* *Trans. Inst. M.E.*, lxii. 46.

that indicators and tachometers can be fitted for testing and should have an efficient governor for speed control.

The fan may be driven by steam, electricity, or compressed air, and in rare cases it is driven through gearing from water-power or turbines.

The usual drives in Great Britain are by—

(i) Direct vertical steam-engine.
(ii) Direct horizontal steam-engine.
(iii) Direct electrical machine.
(iv) Belt or chain drive from steam-engine or electric motor.
(v) Rope drive from steam-engine or electric motor.

Direct drives are the most efficient since there are no transmission losses apart from friction. They show no slip losses.

(i) **The Direct Vertical Steam-engine Drive.**—Direct engines are often used and are usually coupled to the fan shaft by means of a flexible coupling to allow for wear, settling, etc. The vertical engines of the type supplied by Willans, Belliss & Morcom, Browett & Lindley, etc., are very popular at collieries. Vertical engines, though formerly viewed with disfavour, are now in fair demand for the modern high-speed fan. Their rotational speeds are classed as high when over 200 r.p.m., medium when between 100 and 200, and low when below 100. All the light fans of small diameter, including the improved Waddle (see fig. 120), run at speeds approaching 250 to 350 r.p.m.

The usual direct vertical drives are by short-stroke, double-compound engines; their success depends a great deal on forced lubrication. They have the following advantages:—

(*a*) More even crank-pin effort than single engines.
(*b*) Less first cost than similar horizontal engines.
(*c*) Less ground-space occupied.
(*d*) Owing to the high rotational speed, less condensation losses, also less leakage, and usually a better expansive action than the horizontal type.
(*e*) They are smaller for the same duty.

The disadvantages of direct vertical fan engines are—

(*a*) The whole of the pressure, weight of the pistons, etc., comes on to the crosshead pin, which is also stressed throughout the stroke in the double-acting engine.
(*b*) They are not so easy to keep clean and are not under the same visual supervision as the horizontal engine.
(*c*) Vibration is more probable owing to the smaller ground-space occupied.

(ii) **Direct Horizontal Steam-engine Drives.**—These engines were formerly entirely used for the slow-running fans of large diameter which are still to be found in considerable numbers, especially at the older mines. With the many economy devices in respect to expansion, etc., which are now being adopted,

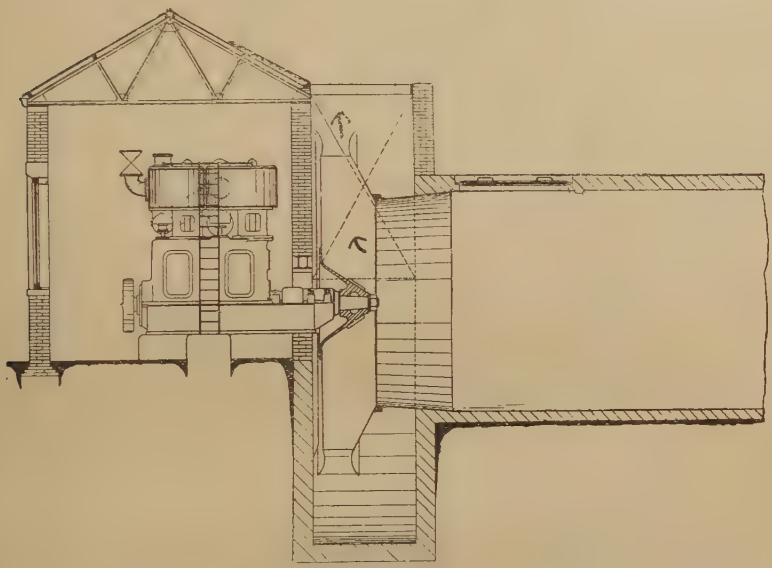


FIG. 120.—Waddle self-contained fan and Belliss engine.

they counteract many of their wasteful defects. Their advantages are—

(*a*) Less liability to vibration than the vertical type, owing to the stresses being distributed over a bigger ground-space.

(*b*) Suited for slow speeds.

(*c*) The moving parts are all under better supervision and more accessible than those of the vertical engine.

The disadvantages of direct horizontal engines are—

(*a*) Heavy pistons need a back-end rod to prevent ovality of cylinder due to wear.

(*b*) They need very heavy foundations, especially if the stroke is long, otherwise the engine will settle and put the fan out of alignment.

(*c*) There is a greater possibility of cylinder condensation.

(iii) **Direct Electric Motor Drives.**—These may be either alter-

nating current (A.C.) or direct current (D.C.), and are usually about 80 per cent. efficient. Owing to the low speed direct-coupled motors are usually bulky and need special attachments, which add to their cost. The chief application of electrical driving for fans is through the intervention of belting, ropes, etc.

Of the D.C. motors the series-wound type are not used owing to the falling-off in speed on increasing the load. For uniform load under fluctuating voltage shunt-wound D.C. motors are used, but where voltage changes are great and sudden compound-wound motors are in favour. D.C. motors are very popular for fan work under varying loads, like developing mines or boosters on progressing headings, but the opening of separation doors, reversal, and other causes in which the inertia of the combined mass of the fan and the air column is quickly changed sometimes causes sparking in these motors.

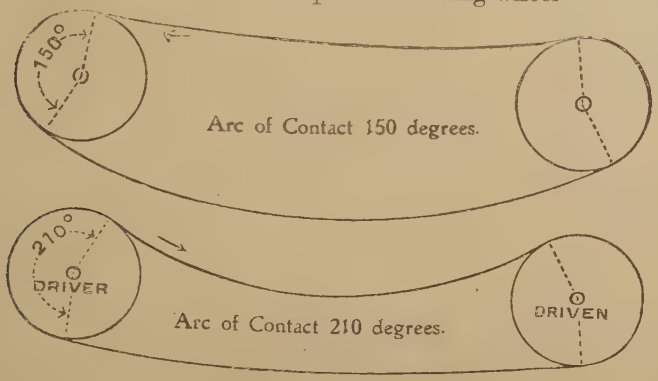
As a general rule A.C. motors find more favour than D.C. They may be of either the induction or synchronous types. Squirrel-cage induction motors have the advantage of wide voltage range and no contact parts in motion, but they consume much current on starting. Wound rotor induction motors are usually arranged for speed regulation by additional means, otherwise they are not efficient, but they do not consume high current on starting. Starting and speed regulation are matters of equal importance with the average colliery fan, so the choice is fairly even between the above two. Synchronous motors are special types which can, when over-excited, be used to keep up the power factor. Though they need to be separately excited by an external D.C. machine, and are more costly, they are efficient in principle and performance.

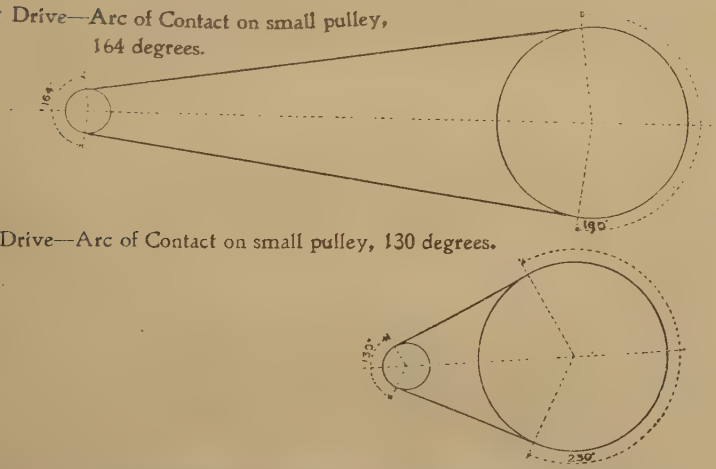
The motor chiefly adopted in Great Britain for fan drives is a lightly-compounded or a shunt-wound motor if D.C. is being used, but if A.C. is used induction motors, either squirrel-cage or slip-ring, are general. These latter types give a fairly-constant speed over a wide range of load variation, are efficient, compact, and trustworthy. They are also capable of long running and considerable overload.

Direct electrical drives are bulky, owing to the low speed, and are therefore costly, often needing gearing. Electrical drives through ropes or belts are very efficient and therefore very popular.

(iv) **Belt or Chain Drives from Steam-engine or Electric Motor.**—Belts made of leather, balata, woven canvas, or other webbed materials are very commonly used at collieries for fan

drives, and usually transmit from 85 to about 90 per cent. of the b.h.p. of the engine to the fan. The maximum speed ratio is about 6 to 1. The relation $\frac{\text{speed of driven wheel}}{\text{speed of driving wheel}}$ is called the





FIGS. 121 and 122.—Showing two methods of increasing the arc of contact.

speed ratio: it is equal to $\frac{\text{diameter of driver}}{\text{diameter of follower}}$. When the frictional grip of the belt is insufficient the driver may slip forward or the belt may slip forward over the follower; this feature is known as slip. "Creeping" of the belt or "creep" is due to unequal stretching of the belt. The belt should be tight on the

under side and slack on the top side, so as to increase the arc of contact (see fig. 121) between belt and wheel, otherwise the shaft centre distance must be changed (see fig. 122). If T_1 in lbs. is the tension on the tight side and T_2 that on the slack side, a belt moving at V ft. per minute will transmit $(T_1 - T_2)$ V ft.-lbs. per minute, or

$$\text{h.p.} = \frac{(T_1 - T_2)V}{33000} \quad . \quad . \quad . \quad (146)$$

Open belts, *i.e.* those in which the driver and follower move in the same direction, are always used on fans. It is not advisable to run belts above 2500 ft. per minute on the smaller pulleys,* and since the diameters have to be kept down the higher the speed

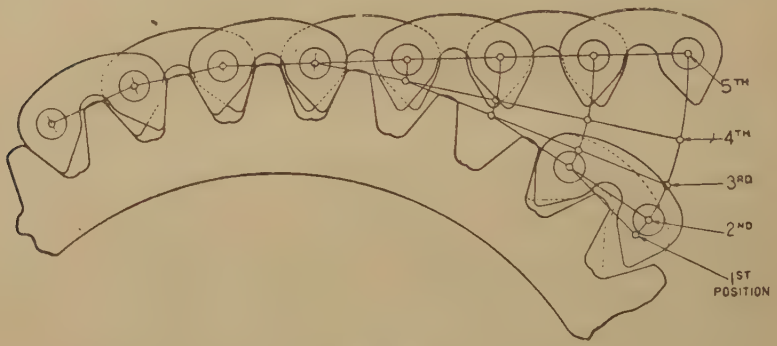


FIG. 123.

the wider the belt to transmit the horse-power. A wider belt means increased stiffness in the belt, which in turn limits the diameter reduction, and has often to be supplemented by perforated or crowned pulleys and other devices to aid the grip. For high speeds and heavy power linked strip belts are used. In fan drives the larger pulley is usually on the fan shaft owing to the comparatively low speed of fans.

Chain Drives.—These may be either silent or bush-roller chains, *i.e.* specially built-up links. They are now becoming popular for fan driving as they save space, have a uniform motion, and economise power.

The Silent Chain consists, as will be seen by fig. 123, of flat plates threaded on studs. The plane tooth faces of the links engage with those of the wheel, and as the chain enters and leaves

* Sargent, *Suction Pumps and Dredgers.*

the wheel with the same peripheral velocity as the wheel-teeth, there is very little impact at entering and no sliding. The result is that the chain runs comparatively quietly and smoothly. The pins which form the joints are of different shapes according to the maker, the main object in the design of them being to render wear, and consequent elongation and alteration of the pitch of the chains, a slow process. The chain lies on the face of the wheel like a belt, and there are several methods corresponding to crowning and flanging a pulley for keeping it in its place.

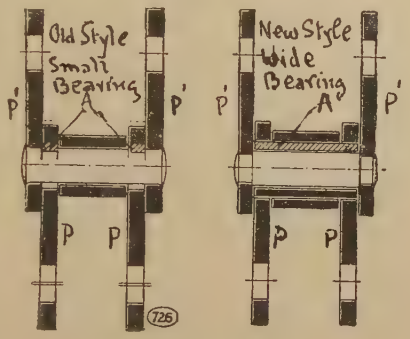


FIG. 124.

The Bush-roller Chain (fig. 124) consists of a series of pin-and-knuckle joints, each having a roller on the outside of the bush to relieve friction between wheel teeth and chain on entering and leaving. The chain consists of inside and outside elements. An inside element is made up of two side plates, P P, with two bushes fixed or keyed on to them and two rollers running loose on the

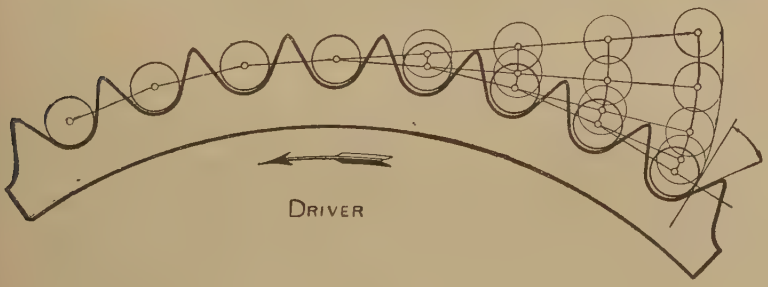


FIG. 125.

bushes. The outside element consists of two side plates, P′ P′, with two pins fixed or keyed into them, and these pins pass through the bushes of the inside element, forming with them the bearing of the chain, see figs. 124 and 125 above.*

* H. T. Hildage, *Transmission of Power by Chains*, and Messrs Hans Renold, Manchester.

The advantages of a Chain Drive are: (i) It is a positive transmission, inasmuch as the chain cannot slip on the wheel as a belt does.

(ii) It is over 98 per cent. efficient, actually.

(iii) In general, the distance between the centre of the driving and driven shafts will be much less than with rope or belt drives. The distance depends on the power to be transmitted and the lie of the drive, *i.e.* horizontal or not.

(iv) Driving chains are capable of running at any mine fan speeds satisfactorily. Drives of small power for auxiliary fans

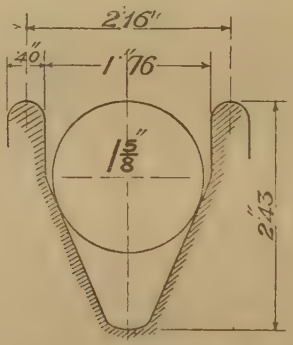


Fig. 126.—Section of groove for cotton driving ropes.

Pitch	. .	equals	1½ times diameter of rope.
Depth	. .	,,	1½ ,, ,, ,,
Width of groove		,,	1½ ,, ,, ,,
Midfeather	.	,,	¼ of diameter.

are commonly used up to 3000 r.p.m., and large drives up to 400 h.p. can be run with the chain up to 500 r.p.m., but higher speed ratios than 6 : 1 shorten the life of the chain.

The disadvantages of Chain Drives are: (i) They require more care and attention than ropes or belts.

(ii) The erection requires to be more accurate. Faults in erection will not cause the chain to run off the wheel but will damage the chain.

(iii) Chains require lubrication, which usually means covering or casing in, in order that the lubricant may not be spread about.

(iv) More care must be exercised, in selecting the most suitable chain for any specific drive, than is needed with ropes or belts.

(v) **Rope Drive from Steam-engine or Electric Motor.**—In this

case the fan is driven by a number of separate cotton ropes, each consisting of three or four strands. Each rope works in specially turned grooves on the follower and driver, the design of which is as shown in fig. 126. The number of ropes varies up to about fourteen, and the diameter to about $2\frac{1}{2}$ in. for ventilating machines. They constitute probably the most popular form of fan drive in this country, being very efficient; over 98 per cent. is commonly attained and seldom below 90 per cent. Rope tensions of over 350 lbs. per sq. in. and speeds up to 5500 ft. per minute are possible. The British standard for rope drives is 200 lbs. per sq. in. tension and 82 ft. per second speed. Very high speeds affect the economy. Two hundred lbs. per sq. in. is considered a reasonable working tension, *i.e.* approximately $160\ D^2$, where D is the rope diameter in inches.*

The most important factor to decide in designing a rope drive is the number and diameter of the ropes required. The power that each rope will transmit depends upon (1) the pulley diameters, (2) the rope speed, and (3) the rope diameter. It is better to overload a small rope than work a large rope over a small pulley, providing always that the slipping point is not reached. Where machinery has to stand idle for a time, such as on changing over to the duplicate fan or during repairs, the rope should be blocked—blocks of wood should be inserted between the rope and the pulley rim. When starting up again the removal of these blocks ensures no hot bearings or similar troubles. The effective tension E on a rope is the total tension T, $(160D^2)$, less the centrifugal tension F. By (35) $F = \frac{WV^2}{g}$ and here W, the weight of one rope in lbs. per ft. is $0{\cdot}27D^2$ * and V is the rope speed in ft. per second.

$$\text{Hence} \qquad E = T - F = 160D^2 - \frac{WV^2}{g} = D^2\left(160 - \frac{0{\cdot}27V^2}{g}\right) \qquad (147)$$

$$\text{Also} \qquad \text{h.p.} = \frac{VE}{550} \qquad (148)$$

E.g. how many 2-in. ropes will be needed to drive a fan taking 380 h.p. from a 12-ft. driving wheel moving at 100 r.p.m. ?

* Thos. Hart, Rope Makers, Blackburn.

$$V = \frac{\pi \times 12 \times 100}{60} = 62{\cdot}83 \text{ ft. per second.}$$

$$W = 0{\cdot}27D^2 = 1{\cdot}08 \text{ lbs. per ft. run.}$$

$$E = T - F = 160D^2 - \frac{1{\cdot}08 \times 62{\cdot}83^2}{32{\cdot}2} = 507{\cdot}6.$$

$$\text{h.p.} = \frac{62{\cdot}83 \times 507{\cdot}6}{550} = 58.$$

Number of ropes required at 2-in. diameter $= \frac{380}{58}$, and adding 33 per cent. for margin of safety, repairs, etc. = 9 ropes.

Advantages of Rope Drives.—(i) Rope driving is very much cheaper in first cost or renewals than electrical installations, belts, or gearing.

(ii) Saving in size of pulleys and driving space; a rope $1\frac{1}{2}$ in. diameter will transmit as much power as a belt 6 in. wide.

(iii) Freedom to drive shafts at almost any distance apart which may be desired and with almost any speed variation.

(iv) Slipping is reduced and power more easily distributed.

(v) Elasticity; adding to the safety of the drive, thereby reducing stoppage.

(vi) Lighter running and less noise.

(vii) Freedom from stoppage or breakdown. If it should happen that a rope gives way at any time it can be easily removed, the remaining ropes doing the work until stopping time or such time as the rope can be repaired or a new one put on in its place.

The following formula will give the length, l, of rope required, distance, C, between shaft centres, size of driver and follower, etc.

$$l = \pi(R + r) + 2\sqrt{C + (R - r)} \quad \begin{cases} R = \text{radius of large pulley} \\ r = \text{,, small ,,} \end{cases} \quad (149)$$

The above applies to an open drive; if the drive is crossed, the sign in the bracketed portion under the root is plus.

A rope drive from an electric motor is considered an ideal arrangement.

The Disadvantages of Rope Drives are bouncing, surging, and mounting.

The causes of bouncing are: (i) Pulleys in faulty alignment, untrue, eccentric, or unbalanced.

(ii) Bad governing or too light a fly-wheel causing speed variation.

(iii) Ropes flapping against something in their flight; this is one of the greatest causes of rope wear.

(iv) Bad splicing: the splice should not be thicker than the normal rope.

Surging is due to heavy fluctuations in the load and is consequently not often met with in mine fan drives.

Mounting is due to the following causes: (i) Slackness; if there is a steady, straight, and even drive, slackness is not a drawback. Ropes should be spliced early to take up slack, but should not be too tight; this causes stretching. Watering a rope to tighten it only causes it to slacken more when dry.

(ii) Rope too thick for the groove: the grooves should be of the dimensions as given in fig. 126.

(iii) Overloading causing slipping; this also tends to lessen the life of the rope.

(iv) Overspeeding: this increases the centrifugal effect, the rope tending to jump its groove. The plucking action of wedged or rusty grooves also causes jumping; a good lubrication should be used.

(v) Harmonic vibration due to the shaft centres being too far apart.

Most of the above defects and their consequences can be avoided by installing a guiding grid or comb, as shown in fig. 127.

The Speed of Fans.—Fan speeds vary considerably with the type. The older machines being of large diameter, attain their peripheral velocity with a small number of revolutions per minute; their diameters vary from about 20 ft. up to 45 ft., and their speeds from about 50 to 150 r.p.m. The peripheral speed of fans is usually from 6000 to 8500 ft. per minute, and the modern type of fan being small—usually from about $7\frac{1}{2}$ to 18 ft. in diameter—maintains this tip velocity by increased revolutionary speed. Modern fan speeds vary from about 120 to 350 r.p.m., the advantage of direct drive is thereby gained.

The velocity, V, of the fan tips may be deduced in the terms with which we are already familiar, and considering a simple fan as follows:—

By (121) *et seq.* $\frac{h}{h_0} = \frac{o^2}{a^2}$

$$\therefore \qquad h_0 = h\frac{a^2}{o^2} \text{ and } h = h_0\frac{o^2}{a^2}$$

$$H = h + h_0 = h + h\frac{a^2}{o^2} = h\left(1 + \frac{a^2}{o^2}\right)$$

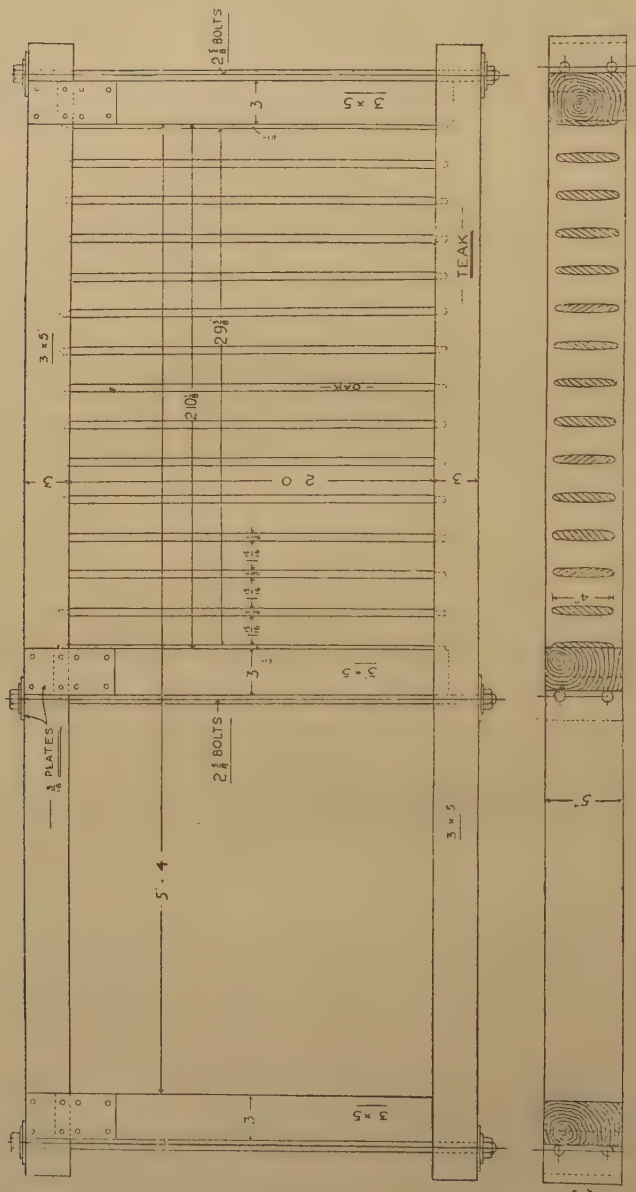


FIG. 127.—Rope comb for fan drive.

By (106) $$h = \frac{H}{\left(1 + \frac{a^2}{o^2}\right)} = \frac{KV^2}{g\left(1 + \frac{a^2}{o^2}\right)}.$$

(In the case of a curved-bladed fan, (247) must be considered here.)

The quantity, Q, in cub. ft. per second.

By (117) $Q = 0{\cdot}65a\sqrt{2gh}$

$$\therefore \quad Q = 0{\cdot}65a\sqrt{\frac{2gKV^2}{g\left(1 + \frac{a^2}{o^2}\right)}} = 0{\cdot}65aV\sqrt{\frac{2K}{\left(1 + \frac{a^2}{o^2}\right)}} \quad . \quad (150)$$

$$\therefore \quad V = \frac{Q}{0{\cdot}65a}\sqrt{\frac{1 + \frac{a^2}{o^2}}{2K}} = \frac{Q}{0{\cdot}65}\sqrt{\frac{a^2+o^2}{2Ka^2o^2}}.$$

Taking quantity in cub. ft. per minute,

$$V = \frac{Q}{0{\cdot}65a60}\sqrt{\frac{a^2+o^2}{2Ka^2o^2}}$$

$$V = \frac{Q}{40}\sqrt{\frac{a^2+o^2}{2Ka^2o^2}} \quad . \quad . \quad . \quad . \quad (151)$$

Knowing the relation between o and a, we may compute the ratio of the mine and total resistances thus—

when $o = 3a$ $$h = \frac{H}{1 + \left(\frac{1}{3}\right)^2} = 0{\cdot}9H.$$

Similarly when $o = 2a$ $h = 0{\cdot}8H$

and when $o = a$ $h = 0{\cdot}5H$.

E.g. (i) Find the relation between the equivalent orifice of the mine and of the fan when the mine resistance is 0·75 times the total resistance.

$$h = \tfrac{3}{4}H = \frac{H}{\frac{4}{3}} = \frac{H}{1 + \left(\frac{\sqrt{1}}{\sqrt{3}}\right)^2} = \frac{H}{1 + \left(\frac{a}{o}\right)^2}$$

$$\therefore \quad o = \sqrt{3} = 1{\cdot}732$$

$$a = \sqrt{1} = 1{\cdot}000$$

E.g. (ii) When $o = 2\frac{1}{2}a$, find the relation between the mine and total resistances.

$$o = 2\tfrac{1}{2}a \quad \therefore \quad h = \frac{H}{1+\left(\frac{1}{2{\cdot}5}\right)^2} = \frac{H}{\frac{29}{25}}$$

$$h = 0{\cdot}862H.$$

Given $o = 2\frac{1}{2}a$, (150) becomes $Q = 0{\cdot}8535aV\sqrt{K}$ . . (152)

and (151) becomes $V = \dfrac{1{\cdot}172Q}{a\sqrt{K}}$ . . . . . (153)

The orifice of passage should be at least twice the orifice of the mine, because when they are equal the fan is producing as much resistance as the mine and is then totally unfitted for its work.

The inlet area should be a little greater than the orifice of passage and not less than twice the equivalent orifice.

By (106) and (119) the resistance of the mine can be regarded as

$$h = \frac{kV^2}{g} - \frac{Q^2}{0{\cdot}65^2o^2 2g} \quad . \quad . \quad . \quad (154)$$

and since the denominator of the latter portion may be considered constant, the fraction can be represented by c, thus

$$h = H - cQ^2 \quad . \quad . \quad . \quad . \quad (155)$$

Therefore for a constant resistance we get—

By (154), depression varies as the square of the fan speed.

By (154), quantity passing varies directly as the fan speed.

The effect of increased tangential velocity is, of course, an all-round benefit to the volume, cooling power of the air, the sparsity of inflammable gases, etc., but *very high velocities have their disadvantages*, among which may be mentioned—

(*a*) The power required increases as the cube of the velocity.

(*b*) If we assume approximately constant temperature and relative humidity, increase of velocity means increased drying power. In a dusty mine this might be very dangerous, especially if there is much blasting.

(*c*) The amount of dust in suspension in the air is increased.

(*d*) Leakage increases with the pressure, which in turn increases as the square of the velocity.

(Leakage in some mines amounts to 80 per cent., *i.e.* only 20 per cent. of the air dealt with by the ventilator ever reaches

the working face. Fifty per cent. would be a fair average leakage for most British coal mines. This figure may be taken as below the average for most metalliferous mines.)

From the foregoing remarks it will be manifest that the speed of the ventilator must be under complete command; this is accomplished by means of the governor. The centrifugal type of governor appears to be most favoured for fan engines, it lends itself to easy setting and variation of throw to meet the "hunting effect" due to varying loads, such as occur in rapidly developing mines and where the air shaft is used for intensive winding.

The importance of a governor can be understood best by noting that constant speed, which can be obtained by efficient governing, also means constant initial water-gauge height (see (106)). If a fall or other obstruction occurs in the airway anywhere between the upcast and downcast the resistance, and therefore the water-gauge reading, is increased for the mine, but the quantity passing, and consequently the power in the air, is diminished. Since there is less quantity passing through the fan, less power will be needed from the fan shaft, so that the fan speed will tend to increase, but the governor preserves a uniform speed. This means that there is a uniform *initial* water gauge, so that if the portion of it taken by the fan be less, the remainder taken by the mine must be greater than that taken before the extra resistance occurred.

Again, if the resistance were relieved by short-circuiting the air, as in the case of the separation doors being opened, the air will encounter only the friction of the shafts and their connecting passage. Very little pressure will be needed to overcome this small resistance and there will be very little difference between the gauge limbs. The quantity of air passing through the fan is increased and the new load tends to slow the fan down; but if the engine is governed, the speed is kept constant, and the power taken from the fan shaft will therefore be increased to cope with the new duty.

These important results may be briefly stated thus—

(*a*) The effect of falls, obstacles, and other resistances is to diminish the quantity passing and the power consumed by the fan, and to increase the visible water gauge and the fan speed.

(*b*) The effect of alleviating the resistance by short-circuiting the air at the separation doors, stripping the upcast casing, etc., is to increase the quantity passing and the power consumed by the fan, and to diminish the visible water gauge and fan speed.

Of course (*a*) and (*b*) are contrary effects due to reversed conditions.

Practical Fan Testing.—The performance of a fan test consists in the first case of the determination of the initial W.G., and since in many cases the closing of the fan drift for this purpose is impracticable, the following method is resorted to :—

We have seen that $\frac{Q}{\sqrt{h}}$ forms a measure of the *mine conductance*, and by Murgue this is equal to $5{\cdot}2 \times a - C$. (See (116) to (118).) Also the *fan conductance* in a similar way $= \frac{Q}{\sqrt{H-h}} = 5{\cdot}2 \times o = c$.

First take a normal reading, Q for the quantity and h the mine pressure, with the fan running at normal speed on the ordinary mine.

Next open the shaft bottom separation doors, so obtaining a greater quantity Q′ and a less head h', since there is only the shaft friction to overcome. This means that C will be greater in the latter case ; but c, which corresponds to the fan friction, does not vary, because the fan speed will be unchanged, so that

$$c = \frac{Q'}{\sqrt{H-h'}} = \frac{Q}{\sqrt{H-h}}$$

whence

$$c = \sqrt{\frac{(Q')^2 - Q^2}{h \quad h'}} \quad . \quad . \quad . \quad . \quad (156)$$

By substitution, H can now be found, and all the remaining data for the test can be read direct from the measuring appliances employed.

With a modern fan the substitution is done with the expression as derived in Chapter X, formula (247).

Fan Sizes.—All mining methods of calculating the size of fans for given conditions are based on the same principles, and in every case the success of the design depends on thoughtful assumptions and the judicious calculation of the resistances expected, and other local factors. Having decided on the lay-out and calculated therefrom the probable maximum resistance due to rubbing surface, etc., for the fully developed mine, an allowance is added for friction in the ventilator, the varying effect of natural ventilation, etc., and the whole expressed as a head or water gauge.

E.g. (i) Suppose we assume that our developed mine will require 300,000 cub. ft. of air per minute to ventilate it efficiently and

that the pressure needed will be $4\frac{1}{2}$-in. W.G. The theoretical water gauge will be about $\frac{7}{8}$ths of this, or $5\frac{5}{8}$ in.; then by (102)—

$$V = \sqrt{\frac{5{\cdot}625 \times 32{\cdot}2 \times 5{\cdot}2}{0{\cdot}0766}} \times 60 = 6654 \text{ ft. per minute.}$$

The next step is to decide on the diameter of the fan, and this depends on the kind of power available. Direct engines, electrical or steam, will, of course, give high peripheral speeds and consequently need only small diameters. Assuming that our conditions will permit of a drive making 200 r.p.m., the diameter of the required fan is—

$$\frac{\text{tip speed}}{\text{circumference}} = \text{r.p.m.} \quad \therefore \quad \frac{6654}{\pi \times 200} = 10\tfrac{1}{2} \text{ ft. approx. diam.}$$

This is a very convenient size for a double inlet forward-bladed fan and should have a volume ratio of 200 per cent., and accepting these conditions the width of such a fan will be—

$$\frac{\text{Quantity}}{\text{fan area} \times \text{speed} \times 2} = \frac{300000}{0{\cdot}7854 \times (10\frac{1}{2})^2 \times 200 \times 2} = 8\tfrac{4}{5} \text{ ft. wide.}$$

The inlet area is got from the equivalent orifice formula, thus—

$$a = \frac{0{\cdot}388 \times 300}{\sqrt{5{\cdot}625}} = 49 \text{ sq. ft.,}$$

and it should be at least twice this amount, as we saw when dealing with fan speeds. This means that we can allow an inlet on one side only of 49×2 or 98 sq. ft. area, or have a double inlet fan, each inlet equal to 49 sq. ft., which gives an inlet diameter of 7·9 ft. or about 16 in. depth of blading, which is very common with the modern types of fan. Collecting our data, we have—

Diameter of fan	10 ft. 6 in.
Width of fan	8 ft. 10 in.
r.p.m. of fan	200.
Manometrical efficiency	80 per cent.
Volumetric ratio	200 per cent.
Inlet diameter	7·9 ft.
Blading depth	16 in.

It will now be seen that the nature of the assumptions or, what is the same thing, one's acquaintance with the requirements of the case in point, decides the calculated sizes of the ventilator. A further few examples will make this clear.

E.g. (ii) A fan exhausting 200,000 cub. ft. per minute at $1\frac{1}{2}$-in. W.G. is to be replaced by another capable of dealing with 300,000 cub. ft., what size fan will be required ?

The new water gauge will be—

$$200^2 : 300^2 :: 1\tfrac{1}{2} : X = 3{\cdot}375\text{-in. W.G.}$$

The speed of blade tips will be—

$$V = \sqrt{2gh} = \sqrt{2 \times 32{\cdot}2 \times 3{\cdot}375 \times 68} = 121{\cdot}6 \text{ ft. per second} = 7296 \text{ ft. per minute.}$$

Assuming the fan to work at 180 r.p.m.

$$\text{Circumference of fan} = \frac{7296}{180} = 40{\cdot}53 \text{ ft. or } 12{\cdot}96 \text{ ft. diameter.}$$

$$\text{Volume discharged per revolution} = \frac{300000}{180} = 1667 \text{ cub. ft.}$$

$$\text{Width of fan} = \frac{1667}{(12{\cdot}96)^2 \times 0{\cdot}7854} = 12{\cdot}82 \text{ ft.}$$

Such a fan can be assumed 75 per cent. efficient, which makes the actual width

$$12{\cdot}62 \times \frac{4}{3} = 16{\cdot}82 \text{ ft.}$$

Summary.—Fan, 13 ft. diam., 17 ft. wide; 180 r.p.m.; clearing 300,000 cub. ft. per minute at a water gauge of $3\frac{3}{8}$ in.

E.g. (iii) *Applying the old method* (note that in all these examples the same root principles hold for the varying methods).

It is required to design a fan to deal with 100,000 cub. ft. against a W.G. of 2 in.

$$\frac{\text{Cubic ft. of water}}{\text{Cubic ft. of air}} = \frac{62{\cdot}3}{0{\cdot}0766} = 815 \text{ in. } \therefore \text{head} = \frac{815 \times 2}{12} = 135{\cdot}8 \text{ ft.}$$

By (101) $V = \sqrt{32{\cdot}2 \times 135{\cdot}8} = 66{\cdot}14$ ft. per second or 3968·4 ft. per minute.

$$\text{Assume 100 r.p.m. then circumference} = \frac{3968{\cdot}4}{100} = 39{\cdot}68 \text{ ft. or}$$

12·63 ft. diam. Allow a width to deal with the quantity over and above the area obstructed by the fan construction, *i.e.* calculate the inlet area with a suitable coefficient.

$$\text{Inlet area by (117)} = a = \frac{0{\cdot}388 \times 100}{\sqrt{2}} = 27{\cdot}44 \text{ sq. ft.}$$

$\therefore$ Inlet diameter = 5·9 ft.

Inlet velocity by $(6a) = \sqrt{2g \times 135{\cdot}8} = 93{\cdot}5$ ft. per second or 5610 ft. per minute. The stated quantity must pass through the inlet, and allowing Murgue's constant we find the check thus—

$$0{\cdot}65 \times 27{\cdot}44 \times 5610 = 100{,}000 \text{ cub. ft. per minute.}$$

If a single inlet fan is being considered, it is customary to allow twice the calculated inlet area on one side. If a double inlet fan is required, the calculated area is allowed on each side. Thus for a double inlet fan—

$$\text{Effective area} = \text{fan side area less the inlet area}$$
$$= 12{\cdot}63^2 \times 0{\cdot}7854 - 27{\cdot}44 = 98{\cdot}0 \text{ sq. ft.}$$

Each foot of width here produces 98·0 cub. ft. and the volume discharged per revolution $= \frac{100000}{100} = 1000$ cub. ft.

$$\therefore \quad \text{Width} = \frac{1000}{98{\cdot}0} = 10{\cdot}2 \text{ ft.}$$

Summary. — Fan, 12½ ft. diameter, 6 ft. inlet, 10 ft. wide, and working at 100 r.p.m.

The weakness of the above method lies in the assumption that the inlet velocity is *always* $\sqrt{2}$ times the tip velocity.

E.g. (iv) *Applying makers' specifications*, which are as follows:—

$$\text{Single inlet fan diameter in inches} = \frac{0{\cdot}5 \times \sqrt{\text{volume}}}{\sqrt[4]{\text{water gauge}}} \quad . \quad (157)$$

$$\text{Double} \quad ,, \quad ,, \quad ,, \quad = \frac{0{\cdot}354 \times \sqrt{\text{volume}}}{\sqrt[4]{\text{water gauge}}} \quad . \quad (158)$$

$$\text{r.p.m.} = \frac{10800\sqrt{\text{water gauge}}}{\text{Diameter in inches}} \quad . \quad . \quad . \quad . \quad . \quad (159)$$

Find the sizes of a double inlet Sirocco fan to exhaust 200,000 cub. ft. per minute against a 3-in. W.G.

$$\text{Diameter in inches} = \frac{0{\cdot}354\sqrt{200000}}{\sqrt[4]{3}} = 120 \text{ in.}$$

$$\text{r.p.m.} = \frac{10800 \times 1{\cdot}732}{120} = 156 \text{ revolutions per minute.}$$

$$\text{h.p.} = \frac{200000 \times 5{\cdot}2 \times 3}{33000} = 94{\cdot}5 \text{ h.p.}$$

Assuming a mechanical efficiency of 75 per cent., the brake horse-power is

$$\text{b.h.p.} = 94{\cdot}5 \times \frac{4}{3} = 126.$$

Summary. — Fan, 120 in. diameter, working at 156 r.p.m., and of 126 b.h.p.

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CHAPTER VII.

THE LAWS OF MINE VENTILATION.

The late J. J. Atkinson embodied the laws governing the motion of fluids as given by Bernoulli, Venturi, D'Arcy, and others, into one compact expression which bears his name. Atkinson's formula—

$$pa = ksv^2 \quad . \quad . \quad . \quad . \quad (160)$$

is then simply a literal representation of the conservation of energy. From it nearly all the laws pertaining to mine ventilation can be deduced.

p = the pressure, usually stated in lbs. per square foot, but often in inches of W.G. or feet of air column. It is usually from 8 to 16 lbs. per square foot in British mines, and its maximum value is about 50 lbs. In certain Continental mines it reaches to about 80 lbs. per square foot. Owing to this figure being so small, when compared with the atmospheric pressure, the air in mines is looked upon as incompressible, or at least the effect of compression due to ventilating pressure is neglected.

a = the cross-sectional area of the roadway, or type airway of the mine, to which any discussion applies. It is always measured in square feet.

k = the coefficient of friction of the air. It is the measure of the pressure expended per square foot of surface traversed in order to create motion under the conditions prevailing. Atkinson placed it at 0 0217 lb. for a velocity of 1000 ft. per minute, and knowing that pressure varies as the square of the velocity this would give $0{\cdot}0217 \times \left(\frac{1}{1000}\right)^2 = 0{\cdot}0000000217$ lb. per square foot of rubbing surface. It can only be found by experiment, using k as isolated from the remaining four terms of (160). It is by no means a constant quantity, hence the reason for the widely divergent results by different researchers that are on record.*

* Lupton's *Mining*, p. 243; Boulton's *Practical Coal Mining*, ii. 286.

There are very acceptable figures—for their particular circumstances—varying from Murgue's $0{\cdot}0017 \times 10^{-6}$ up to $0{\cdot}014 \times 10^{-6}$ by other observers, and even higher. This difference will be easily understood since the friction set up in mines depends on a host of changing conditions, viz. roughness of different materials, changes in direction and temperature, moving bodies, etc., etc. These all tend not only to differ the value of k but also the order of the friction, *i.e.* sliding, rolling, jarring, etc. Furthermore, the ratio of the perimeter to the cross-sectional area, as we have said before, will be a variable resistance, decrease of area creating increase of friction. It is generally considered that k as given by Atkinson is too great, therefore we advise, in the absence of a deduced or given figure, a rough mean of the extremes given above, *i.e.* $0{\cdot}01 \times 10^{-6}$ or 0·00000001. We say $0{\cdot}01^{-6}$ instead of $1^{\ 8}$ because, in order to obviate the tedious repetition of the cyphers, k is usually increased a million times, and the velocity divided by a thousand, whence its square as used in the formula will be lessened a million times, so preserving the original identity. The value of k must be deduced experimentally for every mine where any calculation applies, since we have seen that it may be up to seven or even eight times as great in one place as another. It must always be stated in the same units as p, whether in lbs. per square foot or in inches of W.G. (by dividing by 5·2) or feet of air column $\left(\text{by multiplying the lbs. by 13, } i.e. \ \frac{68}{5{\cdot}2}\right)$. The term "friction factor" has been suggested * instead of "coefficient of friction," because the latter has a distinct and special meaning in mechanics (see Definitions).

s = the rubbing surface in square feet and is obviously the perimeter o of the airway multiplied by the length l, i.e. lo. It constitutes the whole of the area of the mine with which the air comes into contact.

v = the linear velocity of the air; generally stated in feet per minute as distinguished from V, the peripheral speed of the fan tips, which is usually expressed in linear feet per second. Mine velocities may be as high as 2200 ft. per minute in the shafts and up to about 1500 ft. in the main airways. 1000 ft. per minute is a good figure for mains, and 300 to 500 for the faces is

* *Engineering and Mining Journal*, June 1922.

sufficient; the lower figure being the top limit for unshielded lamps, and about 150 the lower limit.

The Origin of Atkinson's Formula.—The more or less approximate laws of fluid friction stated in Chapter I. are largely the result of the investigations of Venturi, D'Arcy, Chezy, and others upon the behaviour of turbulent fluids in circular ducts. Briefly stated, the loss of pressure head, h, due to friction, eddies, bends, junctions, sudden changes in cross-sectional dimensions, etc., has been proved by these experimenters to be—

(i) Independent of the fluid pressure per unit area.

(ii) Nearly proportional to the area of "wetted surface," *i.e.* the surface of the duct actually rubbed in by the fluid. For any full cross-section, a, this is proportional to lp, where l is the length and p the perimeter of the duct.

(iii) Approximately proportional to the square of the velocity, v, of the fluid.

The frictional resistance, F, to the motion of a prism of fluid can, therefore, be considered equal to $cdlpv^2$; d being the weight of unit volume of the fluid and c a suitable constant.

Imagine such a prism of basal area, A, and length, l, moving under the influence of gravity through a distance, s, along a channel of gradient, $i = \frac{h}{l}$ (where h is the vertical height or head causing flow between the two ends of the channel). The energy expended will be $d \times l \times \mathrm{A} \times i \times s$ ft.-lbs. This is equal to the work done against friction, therefore—

$$dl\mathrm{A}is = \mathrm{F}s = cdlpv^2s \quad . \quad . \quad . \quad (161)$$

hence

$$i = \frac{h}{l} = c\frac{p}{\mathrm{A}}v^2 \quad . \quad . \quad . \quad (162)$$

The fraction $\frac{\text{cross-sectional area}}{\text{"wetted" perimeter}} = \frac{\mathrm{A}}{p}$ is known as the "mean hydraulic depth," or m, and is the depth such a volume of fluid would have if it were contained in a rectangular channel with a flat bottom of the same area as the wetted walls of the channel. Therefore the head, h, needed to give a uniform flow, v, along the channel of length, l, and of mean hydraulic depth, m, is, from the above,

$$h = \frac{clv^2}{m} \quad . \quad . \quad . \quad . \quad (163)$$

This is not quite true; the coefficient, c, can only be constant in the case of circular ducts if h is proportional to v^2 and to the wetted perimeter, for all diameters. Later experimenters have proved that these assumptions of Chezy are unwarranted, and several authorities, particularly Prony and D'Arcy, have laid great stress on the degree of roughness of the duct surfaces and sought to express the results of their researches in terms of a binomial function of v.

Professor Osborne Reynolds,* by assuming that the resistance to flow varied with (i) the diameter, length, and roughness of the duct, (ii) the viscosity and density of the fluid, and (iii) the mean velocity of flow, and that it varied with some power of each of these factors, deduced a rational formula which is frequently used and stated as

$$h = \frac{flv^n}{d^x} \quad . \quad . \quad . \quad . \quad . \quad (164)$$

where f is a coefficient deduced experimentally with the values of x and n. Chezy adopted, for practical problems, a form of (163) wherein the coefficient c is written in the form $\frac{f}{2g}$, so obtaining the expression—

$$h = \frac{flv^2}{m2g} \quad . \quad . \quad . \quad . \quad (165)$$

For a given section of circular duct, when deducing the mean hydraulic depth, D the diameter will obviously equal $4m$, so that the above expression is often written thus—

$$h = f\left(\frac{\text{wetted surface}}{\text{cross-sectional area}} \times \text{loss of head due to velocity.}\right)$$

and this becomes ∴

$$h = f\left(\frac{\pi Dl}{\frac{\pi}{4}D^2} \times \frac{v^2}{2g}\right) \quad . \quad (166)$$

$$h = \frac{4flv^2}{2gD}$$

Then if v is given, as is usual, in feet per minute, we get

$$h = \frac{p}{d} = \frac{4flv^2}{2gD3600} \quad . \quad . \quad (167)$$

Where :—

h = pressure drop in feet of head.
f = coefficient of friction from experiment.
l = length of duct (airway) in feet.
D = diameter in feet.
v = velocity of flow in feet per second.
g accel. due to gravity.
d = density of the fluid (air) in lbs. per cub. ft.
a cross-sectional area in square feet.
p = pressure drop in lbs. per square foot.

* *A Dictionary of Applied Physics*, vol. i., Glazebrook.

Multiplying numerator and denominator by $\frac{\pi D}{4}$

$$p = \left(\frac{f}{3600 \times 2g}\right)\frac{dsv^2}{a} \quad . \quad . \quad . \quad (168)$$

(where s is the rubbing surface in square feet).

Now let the bracketed portion $\left(\frac{f}{3600 \times 2g}\right)$ be represented by some new factor, C, then $p = \frac{Cdsv^2}{a}$ . . . . . (169)

and since C and d are constants for any given set of conditions, their product, which we may call k, is also constant, therefore we obtain Atkinson's formula—

$$p = \frac{ksv^2}{a} \text{ as given in (160).}$$

It will be seen that the coefficient of friction k is really a factor C, multiplied by the density d. Now C is a variable depending on the diameter of the duct or airway, the specific gravity of the air, the velocity, the roughness of the surfaces, etc. It cannot be stated in units, and for correct deductions all the above factors on which it depends must be investigated and stated, *and it must be deduced independently for each particular case.*

From the nature of fluid friction one will readily comprehend that Atkinson's formula is only approximately correct, and this chiefly for circular airways. Moreover, the velocity index varies very greatly in ducts of varying roughness. Unwin gives indices varying from 1·75 to over 2 for different stages of roughness, whilst Gibson * gives 1·9 to 1·96 for gravel-lined circular ducts and 2·1 for rectangular rubble masonry sections. (Reference to the authorities cited in the accompanying footnote will assist anyone wishing to pursue this important topic further.) Suffice it to say that the formula, though lacking in fundamental refinement, is certainly the best means for the *practical* investigation of the phenomena of ventilation in the mine.

Modifications of Atkinson's Formula.—The formula, $pa = ksv^2$, is capable of many modifications when taken in conjunction with other ventilation formulæ, and some of the following pages will be devoted to its transformations.

* *Hydraulics*, A. H. Gibson; *Hydraulic Flow Reviewed* (Spon, London), A. A. Barnes.

Dividing (160) throughout by a we get $p = \frac{ksv^2}{a}$. (170)

,, ,, ,, ,, p ,, $a = \frac{ksv^2}{p}$. (171)

,, ,, ,, ,, sv^2 ,, $k = \frac{pa}{sv^2}$. (172)

,, ,, ,, ,, kv^2 ,, $s = \frac{pa}{kv^2}$. (173)

,, ,, ,, ,, ks ,, $v^2 = \frac{pa}{ks}$

whence $v = \sqrt{\frac{pa}{ks}}$. (174)

Squaring (24) and substituting for v^2 in (170) –

$$p = \frac{ksQ^2}{a^3} \quad . \quad . \quad . \quad . \quad (175)$$

Multiplying (160) by v. $pav = ksv^3$ . . . . (176)

This is often written, POWER $= u = ksv^3$ . . . (177)

From cubing (24) $v^3 = \frac{Q^3}{a^3}$ $\therefore$ $u = \frac{ksQ^3}{a^3}$ . . (178)

By considering a circular airway of diameter D (175) transforms from $pa^3 = ksQ^2$

to $p\left(\frac{D^2\pi}{4}\right)^3 = kDl\pi Q^2$

Dividing throughout by πD, $\frac{pD^5\pi^2}{64} = klQ^2$

Whence, since $\frac{\pi^2}{64} = \frac{1}{6{\cdot}5}$, $Q^2 = \frac{pD^5}{6{\cdot}5kl}$

$$\therefore \quad Q = \sqrt{\frac{pD^5}{6{\cdot}5kl}}$$

and $D = \sqrt[5]{\frac{6{\cdot}5klQ^2}{p}}$ ft. . . . (179)

Again (176) may be expressed, with slight modification, as horse-power in the air, thus (where d is the weight of a cubic foot of air and M the motive column):

$$\frac{pav}{33000} = \frac{ksv^3}{33000} = \frac{pQ}{33000} = \frac{MdQ}{33000} = \frac{5{\cdot}2 \text{ W.G.Q}}{33000} \quad (180)$$

The general and most rational deduction of (127) will now be easily grasped (noting the relation between pressure and W.G. head), *i.e.*—

$$p = \frac{ksv^2}{a} = \frac{ks\mathrm{Q}^2}{a^3} \therefore \mathrm{R} = \frac{h}{\mathrm{Q}^2} \text{ or } \mathrm{Q}^2 = \frac{h}{\mathrm{R}},$$

where h is inches of water gauge and R is the constant denoting mine resistance as derived and stated in formula (127), and there defined.

From the foregoing it will be seen that power creates velocity, velocity develops resistance, and resistance demands an equal amount of ventilating pressure.

The Relation between the Intrinsic Resistance of an Airway or Mine and the Power, Pressure, and Volume needed to compensate it.*—Since fixed relations exist between the pressure and power of all airways and their respective areas and rubbing surfaces, as shown in (175) and (176), it will be clear that every airway or mine possesses an implicit resisting faculty; a latent but definite power of resistance dependent on the relation between the area and the rubbing surface. Consequently, for a given airway a given power will set up a definite velocity and evolve a definite resistance, and the ventilating pressure will be equal to, and dependent upon, this resistance. Thus, as stated before, power creates or begets velocity and so evokes resistance in the airway, which, in turn, produces pressure. The velocity and pressure due to any power in an airway, therefore, depends upon the resisting power of the airway; a resisting faculty which the airway possesses latently or inherently, and this is not developed until air flows through the airway. The Americans call this the "potential" of the airway or mine, as the case may be. We prefer a more intimate definition, and will therefore adopt the convenient term "inherence of the airway" when expressed in terms of the area, rubbing surface, etc. Obviously, when stated in terms of the volume, power, and pressure we obtain the "inherence of the air or ventilation," and from the above discussion this must be equal to that of the airway.

From (178) note that Q varies directly as $\sqrt[3]{\frac{a^3}{ks}}$, and by our definition this expresses the *inherent power*, I_u, *of the airway*.

* See "Mine Ventilation," by Tate; "Mine Gases and Ventilation," by Beard; "New Method of Measuring Ventilating Resistances," D. Penman—*Trans. Inst. M.E.*, vol. lxii.

$$\therefore \quad I_u = \frac{a}{\sqrt[3]{ks}} \quad . \quad . \quad . \quad . \quad (181)$$

Again, from (178) Q varies as $\sqrt[3]{u}$, so that the *inherent power of the ventilation* or " in the air " is

$$I_u = \frac{Q}{\sqrt[3]{u}} \quad . \quad . \quad . \quad . \quad (182)$$

which must equal (181), as already defined.

In a similar manner there must be an inherent pressure which will be the relation between the quantity and power and between the quantity and pressure, viz.—

By (175) p varies as Q^2
„ (178) u „ „ Q^3 $\Big\}$ $\therefore \quad I^2{}_p = I^3{}_u$

or $$I_p = I_u \sqrt{I_u} \quad . \quad . \quad . \quad . \quad (183)$$

Whence by (183) and (22) the *inherent pressure of the airway is*

$$I_p = Q\sqrt{\frac{Q}{u}} = \frac{Q}{\sqrt{p}} \quad . \quad . \quad . \quad (184)$$

which again are equal as before.

The inherence, whether of power or pressure, is a primary ratio, therefore for any particular case it is just a number. Its great significance lies in the fact that it is definite for any mine, airway, split, etc., for the conditions existing. Thus it is a radical and correct comparative method which will enable us to deduce the fitness of a ventilator for a particular set of mine conditions without recourse to the doubtful orifice methods.

E.g.—(*a*) What is the inherent power of an airway $7\frac{1}{2}$ ft. $\times$ 8 ft. and 3000 ft. long if 49,000 cub. ft. of air pass at 2 in. W.G. ?

By (181) $$I_u = \frac{a}{\sqrt[3]{ks}} = \frac{60}{\sqrt[3]{\cdot 00000001 \times 93000}} = 614{\cdot}7.$$

(*b*) What is the inherent power " in the air " ?

By (182) $$I_u = \frac{Q}{\sqrt[3]{u}} = \frac{49000}{\sqrt[3]{2 \times 5{\cdot}2 \times 49000}} = 614{\cdot}7.$$

(*c*) What is the inherent pressure ?

By (184) $$I_p = Q\sqrt{\frac{Q}{u}} = \frac{Q}{\sqrt{p}} = \frac{49000}{\sqrt{10{\cdot}4}} = 4712.$$

The inherent power being doubly expressive we can solve in

the alternate form for, say, u or p, and so get the horse-power and unit pressure; and we may also find the effect upon the latter of varying any other of the factors. The root of the whole inherence principle lies in the fact, expressed in (183), that the square root of the pressure varies as the cube root of the power, therefore any ratios of these may be written thus—

$$\left(\frac{p_1}{p_2}\right)^{\frac{1}{2}} = \left(\frac{u_1}{u_2}\right)^{\frac{1}{3}} \text{ or } \sqrt{p_1}\ \sqrt[3]{u_2} = \sqrt{p_2}\ \sqrt[3]{u_1} \quad . \quad (185)$$

and since every mine has an inherent power I_u, and also an inherent pressure I_p, the above relations may be further expressed thus—

$$I_p^2 = I_u^3 = \frac{Q^3}{u} = \frac{Q^2}{p} = \frac{a^3}{ks} \quad . \quad . \quad . \quad . \quad . \quad (186)$$

$$\therefore \quad u = \frac{Q^3}{I_p^2} = \left(\frac{Q}{I_u}\right)^3 = pQ = ks\left(\frac{Q}{a}\right)^3 \quad . \quad . \quad . \quad (187)$$

$$\therefore \quad Q = (uI_p^2)^{\frac{1}{3}} = (pI_u^3)^{\frac{1}{2}} = I_u u^{\frac{1}{3}} = a\left(\frac{u}{ks}\right)^{\frac{1}{3}} = \left(\frac{pa^3}{ks}\right)^{\frac{1}{2}} \quad . \quad (188)$$

$$\therefore \quad p = \left(\frac{Q}{I_p}\right)^2 = \frac{Q^2}{I_u^3} = \frac{u}{Q} = ks\frac{Q^2}{a^3} \quad . \quad . \quad . \quad . \quad (189)$$

$$\therefore \quad s = \frac{a^3}{kI_p^2} = \frac{a^3}{kI_u^3} = \frac{u}{k}\left(\frac{a}{Q}\right)^3 = \frac{a^3 p}{Q^2 k} \quad . \quad . \quad . \quad . \quad (190)$$

From the above the W.G. is got as follows:—

$$\text{W.G.} = \frac{Q^2}{5{\cdot}2I_u^3} \quad . \quad . \quad . \quad . \quad (191)$$

and the horse-power required to produce the ventilation allowing an efficiency of η by

$$\text{h.p.} = \frac{5{\cdot}2 \times \text{W.G.} \times Q}{\eta \times 33000} = \frac{Q^3}{\eta \times 33000 \times I_u^3} \quad . \quad . \quad (192)$$

It will be shown later that the rubbing surface is unaffected by equal splitting, but the total area will, of course, increase in direct proportion as the number of splits because the actual cross-sectional area does not alter nor does the friction factor, k. Consequently, the inherent power varies as the number of splits, n, whilst the inherent pressure varies as the $\frac{3}{2}$ power of the number of splits. Being mindful of the fact that we are dealing with

equal splits and the fundamental notion of inherence, these facts may be stated in the following manner :—

The inherent power of a mine of n equal splits by (181) =

$$I_u = \frac{na}{\sqrt[3]{ks}} \quad . \quad . \quad . \quad . \quad (193)$$

The inherent pressure of a mine of n equal splits by (186) =

$$I_p = na\sqrt{\frac{na}{ks}} \quad . \quad . \quad . \quad . \quad (194)$$

As the portion of the inherent power for the whole system which is obtained in each split will be equal for each, *i.e.* i_u, having deduced it for one split the inherence for the mine will certainly be $ni_u - I_u$, which is equivalent to an additive series composed of the separate inherences of the splits.

The horse-power in such a case as the above will be, by (192)—

$$\text{h.p.} = \frac{Q^3}{\eta 33000(ni_u)^3} \quad . \quad . \quad . \quad (195)$$

If the airways are unequal the inherent power for each differing section must be ascertained by the method of (181), because in such case all the dimensions may be different and only the friction factor cancellable. The horse-power will then be stated, for an airway which has different sections 1, 2, 3, etc., thus—

$$\text{h.p.} = \frac{Q^3}{\eta \times 33000}\left(\frac{1}{i^3_{u1}} + \frac{1}{i^3_{u2}} + \frac{1}{i^3_{u3}} \ldots +\right) \quad . \quad (196)$$

Professor Beard * uses a much more convenient form for this expression by deducing the portion of the inherence (potential) set up in each section without the repeated use of k, which is common, but incorporating it in the final summation. Thus if x_1, x_2, x_3, etc., be the inherent powers of the respective sections as got by omitting the friction factor, k $\left(i.e.\ \text{from}\ x = \frac{a}{\sqrt[3]{s}}\right)$, the following form, easy in application, is obtained :—

$$\text{h.p.} = \frac{kQ^3}{\eta 33000}\left(\frac{1}{x^3_1} + \frac{1}{x^3_2} + \frac{1}{x^3_3} \ldots +\right) \quad . \quad (197)$$

Again, in the case of a continuous mine where the air traverses successive sections of the same airway, the total unit pressure

* J. T. Beard, *Mine Gases and Ventilation* (M'Graw-Hill, U.S.A.).

will be dependent on the sum of the several inherent pressures due to the different sectional dimensions. Calling these sectional inherent pressures i_{p1}, i_{p2}, etc., and the corresponding inherent powers i_{u1}, i_{u2}, etc., we see by (189) that this unit total or mine unit pressure is doubly expressive, thus—

$$p = Q^2\left(\frac{1}{i^2_{p1}} + \frac{1}{i^2_{p2}} + \frac{1}{i^2_{p3}} \dots\right) \text{ or } = Q^2\left(\frac{1}{i^3_{u1}} + \frac{1}{i^3_{u2}} + \frac{1}{i^3_{u3}} \dots\right) \quad (198)$$

Likewise the total unit power here can be, by (187), doubly expressed—

$$u = Q^3\left(\frac{1}{i^3_{u1}} + \frac{1}{i^3_{u2}} + \frac{1}{i^3_{u3}} \dots\right) \text{ or } = Q^3\left(\frac{1}{i^2_{p1}} + \frac{1}{i^2_{p2}} + \frac{1}{i^2_{p3}} \dots\right) \quad (199)$$

Example on (170).—What is the water gauge required to ventilate a mine one mile long and 7 ft. square in cross-section if the velocity is 900 ft. per minute ?

$$p = \frac{ksv^2}{a} = \frac{\overbrace{0{\cdot}01}^{(1000000\times k)} \times \overbrace{28\times 5280}^{s} \times \overbrace{0{\cdot}9\times 0{\cdot}9}^{\left(\frac{v}{1000}\right)^2}}{\underbrace{49}_{a}} = 24{\cdot}4 \text{ lbs. per sq. ft.}$$

$$\frac{p}{5{\cdot}2} = \frac{24{\cdot}4}{5{\cdot}2} = 4{\cdot}7 \text{ in. W.G.}$$

Example on (171).—If a velocity of 1000 ft. per minute is maintained in a mine airway 1100 yds. long when the W.G. reads $3\frac{1}{2}$ and the perimeter of the airway is 36 ft., find the area.

$$a = \frac{ksv^2}{p} = \frac{0{\cdot}01\times 36\times 3300\times 1}{3{\cdot}5\times 5{\cdot}2} = 65{\cdot}27 \text{ sq. ft.}$$

Example on (172).—To produce a velocity of 1300 ft. per minute in an airway 1500 ft. long and 9 ft. square a pressure of $1\frac{1}{2}$ in. W.G. is needed. Find the coefficient of friction.

$$k = \frac{pa}{sv^2} = \frac{1{\cdot}5\times 5{\cdot}2\times 81}{1500\times 36\times 1300\times 1300} = 0{\cdot}000000007 \text{ lb.}$$

Example on (173).—A shaft 20 ft. in diameter absorbs a pressure of $\frac{3}{4}$ in. W.G. in passing 400,000 cub. ft. of air per minute. Find the depth of the shaft.

$$v = \frac{Q}{a} = \frac{400000}{314{\cdot}16} = 1273{\cdot}5 \text{ per minute.}$$

$$s = lo = \frac{pa}{kv^2} = \frac{0{\cdot}75 \times 5{\cdot}2 \times 314{\cdot}16}{0{\cdot}01 \times 1{\cdot}2735 \times 1{\cdot}2735} = 75{,}555 \text{ sq. ft.}$$

$$\therefore \quad l = \frac{s}{o} = \frac{75555}{\pi \times 20} = 1020 \text{ ft. deep.}$$

Example on (174).—What is the velocity needed to produce a $2\frac{1}{2}$ in. W.G. in a mine 1100 yds. long and 6 ft. $\times$ 7 ft. in cross-section ?

$$v = \sqrt{\frac{pa}{ks}} = \sqrt{\frac{13 \times 42}{0{\cdot}01 \times 26 \times 3300}} = 0{\cdot}7975 \text{ thousands, } i.e. \text{ 797·5 ft. per minute.}$$

Example on (176).—Find the horse-power of ventilation when 60,000 cub. ft. of air per minute pass through a mine 6000 ft. in length and $7\frac{1}{2}$ ft. by 8 ft. in sectional area.

$$\text{h.p.} = \frac{u}{33000} = \frac{0{\cdot}00000001 \times 31 \times 6000 \times 1000^3}{33000} = 56{\cdot}37 \text{ h.p.}$$

Example on (179).—A circular shaft 2000 ft. deep passes 250,000 cub. ft. of air per minute and absorbs a W.G. of 1 in. What is the diameter of the shaft ?

$$D = \sqrt[5]{\frac{6{\cdot}5klQ^2}{p}} \qquad \sqrt[5]{\frac{6{\cdot}5 \times 0{\cdot}01 \times 2000 \times 250^2}{5{\cdot}2}} = 17{\cdot}33 \text{ ft.}$$

Example on (180).—200,000 cub. ft. of air circulate at $2\frac{1}{2}$ in. W.G. in a mine. Find the horse-power in the air.

$$\frac{pav}{33000} = \frac{5{\cdot}2 \text{ W.G.Q}}{33000} = \frac{5{\cdot}2 \times 2{\cdot}5 \times 200000}{33000} = 78{\cdot}8 \text{ h.p.}$$

Problems on Inherence.— 1. What is the power and pressure needed to pass 40,000 cub. ft. of air per minute through an airway 8 ft. by $7\frac{1}{2}$ ft. and 3000 ft. long ?

By (181) $I_u = \dfrac{60}{\sqrt[3]{1^{-8} \times 31 \times 2000}} = 687{\cdot}6$

,, (182) $u = \left(\dfrac{40000}{687{\cdot}6}\right)^3$ = 196,800 ft.-lbs. (say 6 h.p.)

,, (189) $p = \dfrac{(40000)^2}{(687{\cdot}6)^3}$ = 4.92 lbs. per sq. ft. (nearly 1 in. W.G.)

2. What would be the effect on the quantity in the above case if the

(*a*) Horse-power was doubled ?
(*b*) Pressure was doubled ?

(*a*) By (188) $Q = 687{\cdot}6(2 \times 196800)^{\frac{1}{3}} = 50{,}400$ cub. ft. per min. (*i.e.* $40000 \times \sqrt[3]{2}$).

(*b*) „ (188) $Q = (2 \times 4{\cdot}92 \cdot (687{\cdot}6)^3)^{\frac{1}{3}} = 56{,}560$ cub. ft. per min. (*i.e.* $40000 \times \sqrt{2}$).

3. A mine has four equal pit-bottom splits, each 9 ft. square and 4000 ft. long (*i.e.* total length 16,000 ft.). If 200,000 cub. ft. per minute enter the mine, find—

(*a*) The pressure at the splits.
(*b*) The water gauge.
(*c*) The horse-power required.

By (193) $I_u = \sqrt[3]{\dfrac{(4 \times 81)^3}{1{\cdot}8 \times 36 \times 16000}} = 180{\cdot}75.$

„ (189) $p = \dfrac{(200000)^2}{(180{\cdot}75)^3} = 6{\cdot}76$ lbs. per sq. ft. $= 1{\cdot}3$ in. W.G.

„ (187) h.p. $= \dfrac{1}{33000}\left(\dfrac{Q}{I_u}\right)^3 = 41$ h.p.

Ans.

3*a*. Check No. 3, above, by reference to the inherent pressure.

By (194) $I_p = \sqrt[3]{\dfrac{(4 \times 81)^3}{1{\cdot}8 \times 36 \times 16000}} = 76{,}810.$

„ (189) $p = \left(\dfrac{200000}{76810}\right)^2 = 6{\cdot}76$ lbs. per sq. ft. $= 1{\cdot}3$ in. W.G.

„ (187) h.p. $= \dfrac{(200000)^3}{33000 \times (76810)^2} = 41$ h.p.

Ans.

4. A mine consisting of two circular shafts 14 ft. in diameter and 1000 ft. deep connected by main airways, intake and return, each 10 ft. by 8 ft. and 2000 ft. long and joining up workings half a mile in extent, of mean cross-section 6 ft. by 6 ft., requires 50,000 cub. ft. of air per minute for its efficient ventilation. Find the horse-power required of the fan which can be taken as 70 per cent. efficient.

By (186), neglecting k, $\dfrac{1}{I_p^2} = \dfrac{s}{a^3}$.

$\therefore$ For the two shafts $\quad \frac{1}{I_p^2} = \frac{2 \times 44000}{(154)^3} = 0{\cdot}0241.$

$\therefore$,, ,, ,, mains ,, $= \frac{2 \times 72000}{80^3} = 0{\cdot}2812.$

$\therefore$,, ,, inbye workings ,, $= \frac{63360}{36^3} = 1{\cdot}3580.$

$\therefore$,, ,, whole mine ,, $= 1{\cdot}6633.$

$$\frac{1}{I_p^2} = 1{\cdot}6633 \quad \therefore \quad I_p = 0{\cdot}7753.$$

By (197) and (187)

$$\text{h.p.} = \frac{k Q^3}{\eta 33000 I^2_p} = \frac{1{\cdot}8 \times (50000)^3}{0{\cdot}7 \times 33000(\cdot 7753)^2} = 90 \text{ h.p.}$$

(This illustrates the enormous power required to pass the air around the workings in a continuous current. By applying (189)

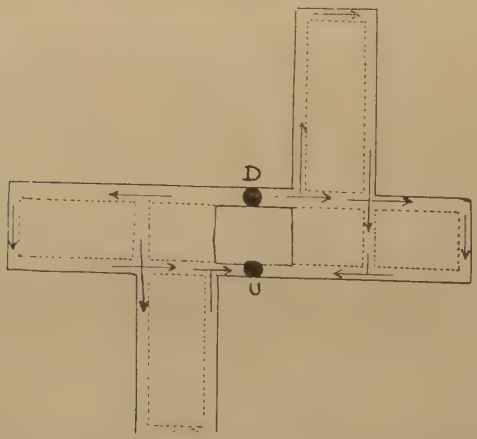

FIG. 128.—Splits.

the water-gauge, which is also very high, will be obtained. Note that the above quantity is small and the conditions very satisfactory.)

Splits.—A split is a division in the ventilating circuit, the current splitting at the point of division and apportioning itself into the several splits according to their resistances, those splits which have least resistance taking most air and *vice versa*. When the distribution is unaided the splits are said to be natural, but

when a resistance is put in any one or more splits, so as to increase the quantity in another split or splits, the apportionment is said to be regulated.

The first or main splits at or near the pit bottom are known as primary ones, and a further subdivision of any of these branches inbye is called a secondary split. It will be understood that the nature of the roadway surfaces is assumed similar throughout. When the splits are equal in area, length, etc., *i.e.* when they divide up the current equally, they are said to be equal splits as distinguished from an erratic apportionment due to unequal resistances. This important fact is so often overlooked by students that a further definition appears to be necessary at this point.

By "equal splits" is meant a division of the original airway into a number of similar airways of the same area each and the same combined rubbing surface, the latter being obtained by curtailing the lengths in the inverse ratio of the number of splits. Thus if there are n equal splits, the length of each will be $\frac{1}{n}$th that of the original airway; or they may be considered as one airway n times the original in area, but of the same rubbing surface. Consequently, by (175), since the rubbing surface, s, is common, the quantity of air, X, passing into n equal splits from a main of cross-sectional area, a, is found thus—

$$\sqrt{a^3} : \sqrt{(na)^3} :: \mathrm{Q} : \mathrm{X}$$
$$i.e. \quad \mathrm{X} = \mathrm{Q}n^{\frac{3}{2}} \qquad . \qquad . \qquad . \qquad (200)$$

In the case of all splits, irrespective of their number at a point, the unit pressure at that point is common for all the splits, and we may also take it that the unit work here is also common to all the splits. That is to say, that $\frac{\mathrm{P}}{a} = \frac{pa}{a} = p$, and $\frac{u}{a} = \frac{pav}{a} = pv$, are equal for each split at this point. These may be expressed in a more familiar manner by reference to (175) and (178), thus—

$$\text{unit pressure} = \frac{pa}{a} = \frac{ks\mathrm{Q}^2}{a^3}; \quad \text{unit work} = \frac{u}{a} = \frac{ks\mathrm{Q}^3}{a^4}$$

and in such case the quantity may be conveniently expressed in terms of the linear dimensions of the airway thus—

For constant unit pressure Q varies as $\sqrt{\frac{a^3}{s}}$ that is as $a\sqrt{\frac{a}{s}}$ (201)

For constant unit work Q varies as $\sqrt[3]{\frac{a^4}{s}}$ that is as $a\sqrt[3]{\frac{a}{s}}$ (202)

These are often denoted by R, *i.e.* (the relative quantity).

E.g. (i) There are three splits at a point in a mine which pass 100,000 cub. ft. per minute. Find the relative quantities for the splits if—

Split A is 7½ ft. × 8 ft. and 2000 ft. long.
,, B ,, 8 ft. × 6 ft. ,, 2500 ft. ,,
,, C ,, 6 ft. × 6 ft. ,, 3000 ft. ,,

The ratio of the total passing into A is $R=\sqrt{\frac{a^3}{s}}=\sqrt{\frac{60^3}{31\times2000}}=1{\cdot}867$

,, ,, ,, ,, B ,, $=\sqrt{\frac{48^3}{28\times2500}}=1{\cdot}257$

,, ,, ,, ,, C ,, $=\sqrt{\frac{36^3}{24\times3000}}=0{\cdot}805$

Total 3·929

The actual quantity ventilating A $=\frac{1{\cdot}867}{3{\cdot}929}\times\frac{100000}{1}=$ 47,510

,, ,, ,, B $=\frac{1{\cdot}257\times100000}{3{\cdot}929}=$ 31,990

,, ,, ,, C $=\frac{0{\cdot}805\times100000}{3{\cdot}929}=$ 20,500

100,000

Note that the greater quantity enters the split of least rubbing surface, *i.e.* A. Other things being equal, the shorter the split the more air it passes, and the greater the need to regulate it for the benefit of the others.

The ratios can be considerably simplified if there exist components of the area or rubbing surface which are common to all the splits. Thus if the lengths are all equal, and as $s = lo$ the component l may be neglected and (201) becomes

$$R \quad \sqrt{\frac{\text{area cubed}}{\text{perimeter}}}=\sqrt{\frac{a^3}{o}} \qquad . \quad . \quad (203)$$

E.g. (ii) Three splits, *a*, *b*, and *c*, of equal length are respectively 7½ ft. by 6 ft., 8 ft. by 7 ft., and 7 ft. by 7 ft. in area, find the relative quantity for each when the total quantity is 200,000 cub. ft. per minute.

For (*a*), $R = \sqrt{\frac{45^3}{27}} = 58{\cdot}1$ and the quantity $\frac{58{\cdot}1}{199{\cdot}4} \times 200000 = 58{,}270$

„ (*b*) „ $\sqrt{\frac{56^3}{30}} = 76{\cdot}5$ „ „ $\frac{76{\cdot}5}{199{\cdot}4} \times 200000 = 76{,}720$

„ (*c*) „ $\sqrt{\frac{49^3}{28}} = 64{\cdot}8$ „ „ $\frac{64{\cdot}8}{199{\cdot}4} \times 200000 = 65{,}010$

Total $= 199{\cdot}4$ Total quantity $= 200{,}000$

In the same manner, if the areas are all similar, the numerator may be replaced by unity, which does not alter on cubing, and the denominator by l since o, the perimeter, is common.

E.g. (iii) Three splits, x, y, and z, all of similar cross-sectional area, are of lengths, 2000, 3000, and 4000 feet respectively. What is the quantity passing into each out of a total of 90,000 cub. ft. in the main ?

Areas, perimeters, and thousands being common,

$$R = \sqrt{\frac{1}{l}} \quad . \quad . \quad . \quad . \quad (204)$$

$\therefore$ for (*x*) $R = \sqrt{\frac{1}{2}} = 0{\cdot}7071$ and quantity $= \frac{0{\cdot}7071}{1{\cdot}7843} \times \frac{90000}{1} = 35{,}670$

„ (*y*) „ $\sqrt{\frac{1}{3}} = 0{\cdot}5772$ „ „ $= \frac{0{\cdot}5772}{1{\cdot}7843} \times \frac{90000}{1} = 29{,}120$

„ (*z*) „ $\sqrt{\frac{1}{4}} = \frac{0{\cdot}5}{\ }$ „ „ $= \frac{0{\cdot}5}{1{\cdot}7843} \times \frac{90000}{1} = 25{,}210$

Total $= 1{\cdot}7843$ Total quantity $= 90{,}000$

Practically speaking the fan speed is not affected by splitting, so that it may be assumed that the main trunk power, U, is constant.* Thus if we have X splits at, or near, the shaft whose total area is Xa, we obtain by (178)

$$U = \frac{ksQ^3}{(Xa)^3} \quad . \quad . \quad . \quad . \quad . \quad (205)$$

Power in the air, friction, rubbing surface, and area being considered constant, it will be seen that quantity, Q, is directly proportional to the number of splits, X, provided they are at the pit bottom, or near it, so that the resistance of the short main leading to them is negligible. If, however, the splits are well inbye, the above ratio does not hold, because the splitting lessens the power at the point of division so that the trunk velocity and

* See Beard's *Mine Gases and Ventilation*, p. 243 *et seq.* Halbaum, *The Regulator, etc.*

resistance will be affected owing to the rubbing surface. The gain in quantity here is therefore less than if the splits were well outbye. The effect on velocity by instituting splits anywhere will be best seen in the following manner. Let there be a number of splits at a certain point in the main ; also let the velocity be v_1 in the main and v_2 in the splits. If the ratio of the total areas a_2 of the splits, to a_1 that of the main, be X, and the ratio of the rubbing surfaces s_2 in the splits, to s_1 that in the main, be Y, then

$$s_2 = \mathrm{Y}s_1 \text{ and } a_2 = \mathrm{X}a_1 \quad . \quad . \quad . \quad (206)$$

and for a stated quantity $$v_2 = \frac{v_1}{\mathrm{X}} \quad . \quad . \quad . \quad . \quad (207)$$

and as the total power u is the sum of the power u_1 absorbed in the main and u_2 in the splits

$$u = u_1 + u_2 \quad . \quad . \quad . \quad . \quad (208)$$

which is by (176) $u = ks_1v_1^3 + ks_2v_2^3$, $= k(s_1v_1^3 + s_2v_2^3)$. (209)

Substituting the value of s_2 from (206) and v_2 from (207), (209) becomes

$$u = ks_1v_1^3\left(1 + \frac{\mathrm{Y}}{\mathrm{X}^3}\right) \quad . \quad . \quad . \quad (210)$$

or $$u = ks_1v_1^3\frac{(\mathrm{X}^3 + \mathrm{Y})}{\mathrm{X}^3} \quad . \quad . \quad . \quad (211)$$

From this, if the power be taken as constant, any decrease in the bracketed term increases v_1, so that the velocity in the main, *i.e.* shaft, or main airway or both, is proportional to the reciprocal of the enclosed term or

$$v_1 \propto \sqrt[3]{\frac{\mathrm{X}^3}{\mathrm{X}^3 + \mathrm{Y}}}, \text{ *i.e.* } v_1 \text{ varies as } \frac{\mathrm{X}}{\sqrt[3]{\mathrm{X}^3 + \mathrm{Y}}} \quad . \quad (212)$$

In a similar manner, from (207) on, we find that the velocity in the splits, v_2, varies as

$$\frac{1}{\sqrt[3]{\mathrm{X}^3 + \mathrm{Y}}} \quad . \quad . \quad . \quad . \quad (213)$$

These last two formulæ show that the main velocity is increased as the ratio of the areas X increases, and that in the splits decreased. The quantity passing depends directly on v_1, so we may also say that—

Total quantity passing varies as

$$\frac{X}{\sqrt[3]{X^3+Y}} \quad . \quad . \quad . \quad . \quad (214)$$

The *total* resistance R = P is the sum of that developed in the main and in the splits and by (170) is

$$P = ks_1v_1^2 + ks_2v_2^2$$

Substituting s_2 from (206) and v_2 from (207)

$$P = ks_1v_1^2\left(1+\frac{Y}{X^2}\right)$$

$$= ks_1v_1^2\left(\frac{X^2+Y}{X^2}\right) \quad . \quad . \quad . \quad (215)$$

wherefore by (212) and (215) since k and s_1 can be left

$$P \propto \left(\frac{X}{\sqrt[3]{X^3+Y}}\right)^2 \times \frac{X^2+Y}{X^2}$$

or the total resistance varies as

$$\frac{X^2+Y}{\sqrt[3]{(X^3+Y)^2}} \quad . \quad . \quad . \quad . \quad (216)$$

E.g. (iv) The main quantity and velocity at a mine are 200,000 cub. ft. per minute and 1000 ft. per minute respectively. There are three equal splits near the pit bottom, each $\frac{3}{4}$ the area of the main and eight times its rubbing surface. What is the effect on the quantity and velocity in the main if another similar split be added ?

First ratio $X = 3\times\frac{3}{4} = \frac{9}{4}$	Second ratio $X_1 = 4\times\frac{3}{4} = 3$
,, ,, $Y = 3\times 8 = 24$	,, ,, $Y_1 = 4\times 8 = 32$

The ratios for quantity or velocity are, by (214) or (212),

$$\frac{2{\cdot}25}{\sqrt[3]{(2{\cdot}25)^3+24}} : \frac{3}{\sqrt[3]{3^3+32}} :: 1 : 1{\cdot}1245$$

The new quantity $= 200000\times 1{\cdot}1245 = 224900$.

The new velocity in main $= 1000\times 1{\cdot}1245 = 1124{\cdot}5$ ft. per minute.

E.g. (v) The resistance of a mine, which has two equal pit bottom splits, each $\frac{1}{3}$ the shaft area and 6 times its rubbing surface,

is 13 lbs. per sq. ft. What will be the resistance when there are four such similar splits ?

$$X = 2 \times \tfrac{1}{3} = \tfrac{2}{3} \qquad X_1 = 4 \times \tfrac{1}{3} = \tfrac{4}{3}$$
$$Y = 6 \times 2 = 12 \qquad Y_1 = 6 \times 4 = 24$$

By (216)

$$\frac{X^2 - Y}{\sqrt[3]{(X^3 + Y)^2}} : \frac{X_1^2 - Y_1}{\sqrt[3]{(X_1^3 + Y_1)^2}} :: 1 : 1{\cdot}245$$

$$13 \times 1{\cdot}245$$
$$16{\cdot}2 \text{ lbs.}$$

We may now summarise our results as follows :—

Points of Importance regarding Splitting.—I. As the ratio of the total area of the splits to that of the main increases, the main velocity increases (as also does the quantity and the resistances) and the split velocity decreases.

II. By (179) the quantity passing in any split, single airway, or mine, is proportional to the square root of the fifth power of its diameter.

III. The splits should be as near the downcast bottom as possible and the reunions as near the upcast bottom as possible.

IV. The splits should be as nearly equal as possible to make the ventilation in each district proportionate. Also if we have to regulate any split owing to disproportion, the total quantity circulating is reduced thereby, as will be shown later.

V. *In remote parts* of the mine it will be better to install auxiliary fans than resort to regulating any split, for the above reason.

VI. Temperature rise in the air current depends, for a given depth, on the duration of contact with the heated strata surfaces. The splits should be so arranged that they reunite at the same time or nearly so.

VII. The effect on the split velocity given by (213) decides the limit of efficient splitting; beyond this any splitting may be considered excessive, which practice induces high temperatures, humid atmospheres, sluggish currents, bodily discomfort, and perhaps spontaneous combustion.

VIII. Splitting often isolates a portion of a ventilating system, or mine, in case of an explosion; this, of course, depends on the nature of the explosion. Besides increasing the quantity passing, splitting forms a very valuable adjunct to output regulation, uniform advance, and general systematic development.

IX. The aggregate lengths of all the splits does not, in most cases, exceed that of the single tortuous current which would exist if the mine had no splits. In such cases the quantity increases at a greater rate than the number of splits.

X. One of the disadvantages of splitting is that it makes a draw on the trunk or shaft main pressure, which will in turn determine the useful limit of division, as under VII. above.

Regulators.—When investigating the theory of the regulator the student will find the method of negative argument very helpful, *i.e.* a set of apparently true assumptions are adopted and these eliminated by proof until only the truthful residue remains.

The regulator is usually a door or stopping having an adjustable or sliding shutter, by which the area of an aperture in the door can be altered as desired (see fig. 129).

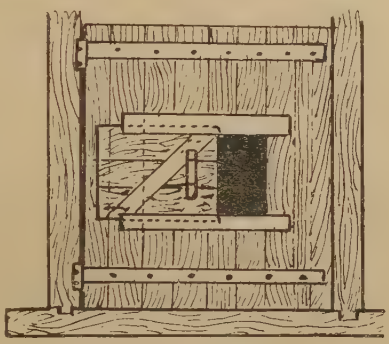

FIG. 129.—A regulator.

The function of a regulator is to redistribute the ventilating pressures ; in doing this it also, in some measure, reapportions the quantities passing in the remaining airways of the mine. The quantity of air cut off or lost by inserting a regulator in any airway is much greater than that gained in the remaining free airways. This fact has brought the regulator somewhat into disrepute, but it will be shown later that as an augmenter of the pressure difference—upon which flow and the clearance of gases really depend—it is certainly useful if wisely applied. We may prove the above statement regarding the waste of quantity due to regulators thus—

E.g. (i) Suppose we have four pit-bottom splits, as is common in many localities, and we are dividing 120,000 cub. ft. of air per minute equally between them, the total ventilating pressure being 4 in. W.G., of which $1\frac{1}{2}$ in. is taken up by the shafts. Now, if we wish to regulate any of these splits the effect on the total pressure may be found by assuming tentatively that the shaft bottom pressure remains unaltered. (This is a customary but dubious assumption to which we will refer hereafter.*) Suppose

* See H. W. Halbaum, *The Natural Philosophy of a Ventilating Regulator.*

we have regulated three of the splits—the procedure is the same, with slight modification, as for only one regulated split—so that 10,000 cub. ft. per minute less passes into each regulated split than into the free one. The latter will then have 30,000 and the others 20,000 each, or a total of 90,000 cub. ft. per minute. Since the shaft area is the same the pressure will vary as the square of the quantity passing, *i.e.* (clearing the cyphers) $12^2 : 9^2 :: 1\frac{1}{2} : x =$ 0·84 in. We thus lose 1·5 − 0·84 = 0·66 in. of pressure in the shafts, and this reduces the total pressure to 4 − 0·66 = 3·34 in. provided we adhere to the erroneous assumption that the pit-bottom pressure is unaltered, which it is not, since the regulator redistributes pressures. The correct effect on the quantities may be found as follows :—

Let Q = the total quantity.
Q′ = the quantity in the free split.
p' = the shaft resistance.
p'' = the split resistance.
k' = the shaft constant.
k'' = the free split constant.

Other things being equal,

$$k' = \frac{Q^2}{p'} = \frac{(120)^2}{1{\cdot}5} = 9600$$

and

$$k'' = \frac{Q'^2}{p''} = \frac{30^2}{4 - 1\frac{1}{2}} = 360$$

Similarly

$$p' : p'' :: \frac{Q^2}{k'} : \frac{Q'^2}{k''} :: 4 \text{ in.} \quad . \quad . \quad . \quad (217)$$

Dealing in thousands for convenience ; we have regulated from 120 to 90, *i.e.* reduced the total by 30 and increased the free split amount by x. Q now becomes—

$$Q = 120 - 30 + x = 90 + x$$

and

$$Q' = 30 + x$$

substituting in (217) $\dfrac{(90+x)^2}{9600} + \dfrac{(30+x)^2}{360} = 4$

$\therefore$ $x = 3{\cdot}37$ (thousands) cub. ft. per minute.

This is an increase for the free split.

The pressures will be in proportion to the square of the quantities and in the shafts will be

$$120^2 : 93{\cdot}37^2 :: 1\tfrac{1}{2} : x = 0{\cdot}9006 \text{ in. W.G.}$$

but in the free split it will be increased since the volume is increased and will be

$$30^2 : 33{\cdot}37^2 :: 2\tfrac{1}{2} : X = 3{\cdot}0924 \text{ in. W.G.}$$

This, by comparison with our former results, illustrates the fallacy of the constant pit-bottom pressure assumption. Collecting our data, we have—

Original quantity	120,000 cub. ft.
Total quantity passing through the regulators . .	60,000 ,,
,, ,, ,, ,, free split. . .	33,370 ,,
,, ,, ,, ,, mine after regulating	93,370 ,,
Quantity gain in the free split	3,370 ,,
,, loss in the main body	26,630 ,,
Ventilating pressure in the shaft	0·9006 in. W.G.
,, ,, lost in shaft	0·5994 in. ,,
,, ,, in the splits	3·0994 in. ,,
,, ,, gain in the splits	0·5994 in. ,,
Total ventilating pressure . . .	4·0000 in. W.G.

It will be seen that in the above case about eight times as much more of the quantity is lost than gained. The increase in pressure is expended, in the regulated splits, at the regulators.

The pressure expended at the regulators in the foregoing case is $3{\cdot}099 - 2{\cdot}5 = 0{\cdot}6$ in., or about 40·8 ft. of air column. The velocity due to this head is by (6*a*) and (117) $= 0{\cdot}65\sqrt{64{\cdot}4 \times 40{\cdot}8}$ $= 33{\cdot}3$ ft. per second, *i.e.* 2000 ft. per minute.

$$\text{Area of regulator} = \frac{\text{quantity}}{\text{velocity}} = \frac{20000}{2000} = 10 \text{ sq. ft. or } 3{\cdot}16 \text{ ft. square.}$$

From the above it will be noted that the amount of pressure which can be taken from the mains decides the performance of a regulator, and it will be shown now that even our assumption that the total pressure is fixed is not quite true. (The foregoing case was taken for convenience in illustration.) The total pressure will require to increase as the distance inbye increases, because that of the main roadway is added to that of the shafts.

E.g. (ii) Suppose that three equal splits are made inbye at such a distance that the pressure for the main road alone is $1\frac{1}{4}$ in. W.G., in addition to the $1\frac{1}{2}$ in. expended in the shafts. The splits having a common pressure of 1 in. pass 40,000 cub. ft. each out of a total of 120,000. It is required to apportion the quantities in two of the splits by regulators in the ratio of 3 : 4, and to raise

the quantity in the remaining free split to 50,000 cub. ft. per minute. Find the effect on quantities and pressures.

The total main pressure $= 1\frac{1}{2} + 1\frac{1}{4} + 1 = 3\frac{3}{4}$ in., and in order to raise the quantity in the free split to 50,000 the common split pressure must be raised to $\left(\frac{50}{40}\right)^2 \times 1 = 1{\cdot}5625$ in. This leaves $3{\cdot}75 - 1{\cdot}5625 = 2{\cdot}1875$ in. for the mains instead of $1\frac{1}{2} + 1\frac{1}{4} = 2{\cdot}75$ in. The quantity passing will now be

$$\sqrt{\frac{2{\cdot}1875}{2{\cdot}75}} \times 120000 = 107{,}100 \text{ cub. ft.}$$

$107{,}100 - 50{,}000$ leaves 57,100 to be shared in the remaining splits in the ratio 3 : 4 by means of regulators, *i.e.* 24,472 in one and in the other 32,628 cub. ft., and these will have pressures of 0·347 and 0·665 inch W.G. This again illustrates that the total quantity is reduced, because the total resistance for a given quantity is increased. It further shows by adding up the pressures that the total mine pressure is increased, and that the assumption of constancy on behalf of any partial or of the total pressure is incorrect. The pressure p, as given by Atkinson's formula, which is required to pass the quantity Q in any airway regardless of regulators, now becomes p_2. p_2 is compounded of the above pressure p, which creates the normal velocity of the airway and of p_1, the pressure needed to create the velocity in the vena contracta of the orifice, *i.e.* p_1, is the difference in pressure on both sides of the regulator and would be read off on the regulator water gauge, thus—

$$p_2 = p + p_1 \quad . \quad . \quad . \quad . \quad (218)$$

It will now be seen that the split formula (201) *as deduced from Atkinson's expression cannot apply directly to the regulator,* and that p_1 can be obtained by means of the equivalent orifice laws, W.G. $= \left(\frac{0{\cdot}38Q}{a}\right)^2$ or $\frac{0{\cdot}15Q^2}{a^2}$, etc. If it is desired to work in pressures expressed as feet of head, then the conversion from lbs. is made by multiplying by 13, since there are $\frac{1}{0{\cdot}0766} = 13$ ft. of air in a column weighing 1 lb. and 1 sq. ft. in cross-section. The relation between this and 1 in. W.G. = 68 ft. approx. will be obvious.

That a regulator may increase the efficacy of a ventilating

current, even though it squanders quantity, will be seen in the case of a gassy mine where its introduction, by increasing the difference of pressure either side of an emission, diminishes the likelihood of further accumulations. Take a simple ideal mine, as in fig. 130, in which a continuous current of 100,000 cub. ft. per minute circulates at a pressure of $1\frac{1}{2}$ in. W.G., say 100 ft. head. Assume also that the whole pressure is uniformly dissi-

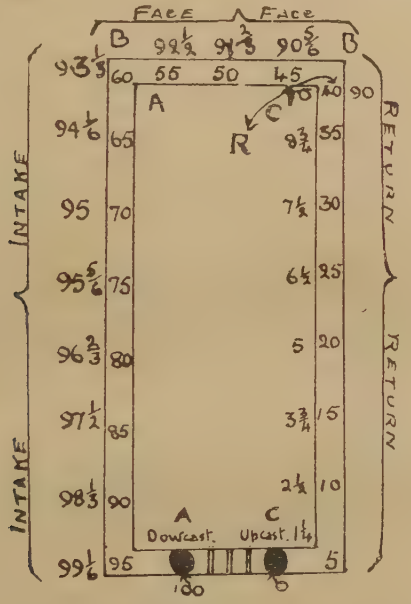

FIG. 130.—Regulator placed at R.

pated in overcoming the mine resistance, and A A is the intake, B B the face, and C C the return. Now if a regulator be placed at R, the beginning of the return behind the last working place, so as to halve the quantity passing, we know by (175) that the pressure will vary as the square of the quantity and will become $\left(\frac{1}{2}\right)^2 = \frac{1}{4}$ or $\frac{40}{4} = 10$ ft. of head, where previously it was 40 ft. On the intake side it will now be $100 - 10 = 90$ ft., and the pressures will reapportion themselves on both sides of the regulator, as shown by the numbers on the inside of fig. 130 for the return, and on the outside for the intake and face; the figures between the lines show the original pressures. By this arrangement we

have increased the intake pressure in some parts of the face by about 100 per cent., so that if any gas exuded in these places previous to inserting the regulator the probability of it doing so now is only half that pertaining before. If the regulator was placed in the intake the pressure on the face would fall as well as in the return, and if it be put in the outbye end of the return the pressure in the return would be increased with that in the intake and on the face, both of which conditions are wasteful and dangerous. *The best place for a regulator is just behind the last working place and on the return side.* It will now be recognised why it is that the effectiveness of a regulator can often be enhanced by placing it farther inbye on the return side as the work advances. In the first of the foregoing examples it was shown that the effect of a regulator is to increase the common pressure to which the whole of the splits are subject. We assumed the total pressure to be the same for convenience in illustrating the quantity effect. The second example contradicts this assumption, and shows, as above stated, that regulation increases the total ventilating pressure. We may further look upon this aspect of the subject in this way; the fan uses up the same amount of energy after as before regulating, so that the number of ft.-lbs., or $p \times Q$, is constant, but Q falls, so p increases (to p_2). The following examples detail the correct procedure. It will be noticed that the shortest splits (other things being equal) take the most air and, consequently, must have the smallest regulator orifice, *i.e.* they must set up the most resistance in order to apportion the quantities properly.

E.g. (iii) It is required to pass 50,000 cub. ft. per minute into each of three splits (*a*), (*b*), and (*c*), whose sizes are—

(*a*) 2000 ft. long and $7\frac{1}{2}$ by 8 ft. in cross-sectional area.
(*b*) 2500 ft. ,, 7 ,, 7 ft. ,, ,, ,,
(*c*) 3000 ft. ,, 6 ,, 8 ft. ,, ,, ,,

Find the sizes of the regulators required in the two splits taking the most air—

The pressure necessary to ventilate the largest split must be maintained, and this will be the common pressure for all the splits. Taking Q in thousands and $k = 0{\cdot}01$ we get, by (175), pressure p for (*c*), the largest split, thus—

$$p = \frac{ksQ^2}{a^3} = \frac{0{\cdot}01 \times 3000 \times 28 \times 50 \times 50}{48 \times 48 \times 48} = 18{\cdot}99 \text{ lbs. per sq. ft.}$$

for (*b*) $p = 11{\cdot}87$ lbs. or a difference of $4{\cdot}12$ lbs. per sq. ft.
and for (*a*) $p = 7{\cdot}17$ lbs. ,, ,, $11{\cdot}82$,, ,,

Now the regulators will be in (*a*) and (*b*), and must be of such area that they will create a resistance equal to their respective pressure differences from that of the common pressure. In inches of water gauge the difference is 2·28 in. for (*a*) and 0·79 in. for (*b*). Hence the area of the orifice of the regulator in (*b*) is—

$$\frac{0{\cdot}388Q}{\sqrt{0{\cdot}79}} = \frac{0{\cdot}388 \times 50}{0{\cdot}89} = 21{\cdot}8 \text{ sq. ft.}$$

and similarly for (*a*) orifice area = 12·85 sq. ft.

E.g. (iv) Suppose all the areas are equal in the splits, then we have a pressure difference in (*b*) and (*a*) proportionate to the length by which each falls short of (*c*), the standard split.

In (*b*) it will be $\frac{(3000-2500)}{3000} = \frac{18{\cdot}99}{5{\cdot}2} = 0{\cdot}6$ in. W.G.

,, (*a*) ,, ,, $\frac{(3000-2000)}{3000} \times \frac{18{\cdot}91}{5{\cdot}2} = 1{\cdot}2$ in. W.G.

and by (117), as before, the regulator aperture area for (*b*) = 25 sq. ft. and for (*a*) = 17·7 sq. ft.

It will now be seen that the pressure, p, lost in its own split by inserting a regulator is, as given by 4*a* and under definition for (94), equal to

$$p = \frac{d}{2g}(v_1^2 - v^2) \quad . \quad (219)$$

Where
- d = the weight of a cub. ft. of air.
- v = the velocity of approach.
- v_1 = the velocity through the aperture.
- g = the acceleration due to gravity.

We are now in a position to summarise the chief laws governing the ventilation of mines. These will be stated together in groups for easy reference, and in every case other things will be considered equal.

THE LAWS OF QUANTITY.

Q1. By (175) quantity varies DIRECTLY as the square root of the pressure, *i.e.* as $\sqrt{p}$.

Q2. ,, (175) ,, ,, ,, square root of the W.G. or head, *i.e.* as $\sqrt{\text{W.G.}}$ or $\sqrt{h}$.

Q3. ,, (117) ,, ,, ,, mine orifice, *i.e.* as a.

Q4. ,, (119) ,, ,, ,, fan orifice, *i.e.* as o.

Q5. ,, (201) ,, ,, ,, square root of the area cubed over the rubbing surface (pressure constant), *i.e.* as $\sqrt{\frac{a^3}{s}}$.

Q6. ,, (152) ,, ,, ,, speed of the fan, *i.e.* as V.

Q7. ,, (179) ,, ,, ,, square root of the fifth power of the airway diameter, *i.e.* as $\sqrt{\text{D}^5}$.

Q8. ,, (152) ,, ,, ,, square root of the manometrical efficiency, *i.e.* as $\sqrt{\text{K}}$.

Q9. ,, (178) ,, ,, ,, cube root of the power, *i.e.* as $\sqrt[3]{u}$.

Q10. ,, (23) ,, ,, ,, area (velocity constant), *i.e.* a.

Q10a ,, (202) ,, ,, ,, cube root of the fourth power of the area over s. (power unit constant), *i.e.* as $\sqrt[3]{\frac{a^4}{s}}$.

Q11. ,, (175) ,, INVERSELY ,, square root of the rubbing surface, *i.e.* as $\sqrt{\frac{1}{s}}$.

Q12. ,, (175) ,, ,, ,, square root of the length or perimeter of the airway, *i.e.* as $\sqrt{\frac{1}{l}}$ or $\sqrt{\frac{1}{o}}$.

Q13. ,, (175) ,, ,, ,, square root of the coefficient of friction, *i.e.* as $\sqrt{\frac{1}{k}}$.

The Application of the Laws of Quantity.

Example on Q1 *and* Q2.—With a water gauge of 2 in., 100,000 cub. ft. of air per minute circulate in a mine. What will the quantity be if the pressure is increased 50 per cent. ?

$$\sqrt{2} : \sqrt{3} :: 100000 : X = 122{,}500 \text{ cub. ft. per minute.}$$

Example on Q3 *and* Q4.—The equivalent orifice of a mine passing 150,000 cub. ft. per minute is 24 ft., what will the quantity be when the orifice is 16 ft. ?

$$24 : 16 :: 150000 : X = 100{,}000 \text{ cub. ft. per minute.}$$

Example on Q5 *and* Q11.—40,000 cub. ft. of air per minute pass through a split 8 ft. wide by $7\frac{1}{2}$ ft. high. If the cross-sectional dimensions be altered to 7 ft. by 7 ft., what will then be the quantity passing, assuming constant pressure ?

$$\sqrt{\frac{60^3}{31}} : \sqrt{\frac{49^3}{28}} :: 40000 : X = 31{,}063 \text{ cub. ft. per minute.}$$

Example on Q6.—A fan making 250 r.p.m. circulates 200,000 cub. ft. of air per minute through a mine: if the speed be increased to 300 r.p.m., what quantity will then in pass the mine ?

$$250 : 300 :: 200000 : X = 240{,}000 \text{ cub. ft. per minute.}$$

Example on Q7.—A circular shaft 15 ft. in diameter admits 120,000 cub. ft. of air per minute into a mine, what quantity would a 21-ft. shaft pass under the same conditions ?

$$\sqrt{15^5} : \sqrt{21^5} :: 120000 : X = 278{,}400 \text{ cub. ft. per minute.}$$

Example on Q8.—When the manometrical efficiency of a ventilator is 70 per cent. it passes 300,000 cub. ft. of air per minute; what will it pass if the efficiency be decreased to 60 per cent. ?

$$\sqrt{0{\cdot}7} : \sqrt{0{\cdot}6} :: 300000 : X = 277{,}750 \text{ cub. ft. per minute.}$$

Example on Q9.—If 150,000 cub. ft. of air pass per minute when the horse-power is 150, what is the effect on the quantity of increasing the power to 200 ?

$$\sqrt[3]{150} : \sqrt[3]{200} :: 150000 : X = 165{,}100 \text{ cub. ft. per minute.}$$

Example on Q10*a*.—40,000 cub. ft. of air pass in an airway $7\frac{1}{2}$ ft. by 8 ft. and 2000 ft. long; if the airway be reconstructed

to 9 ft. square and 3000 ft. long and the power unit maintained constant, what will be the quantity under the new conditions ?

$$60\sqrt[3]{\frac{60}{62000}} : 81\sqrt[3]{\frac{81}{108000}} :: 40000 : X = 49{,}610 \text{ cub. ft.}$$

Example on Q5, Q11, *and* Q12.—A roadway 1500 ft. long and 7 by 8 ft. in cross-sectional dimensions passes 60,000 cub. ft. of air per minute; what quantity will pass through it when

(*a*) The length is doubled ?
(*b*) The height is 9 ft. and the width 10 ft. ?

(*a*) $\frac{1}{\sqrt{1}} : \frac{1}{\sqrt{2}} :: 60000 : X = 42{,}430$ cub. ft. per minute.

(*b*) $\sqrt{\frac{56^3}{30}} : \sqrt{\frac{90^3}{38}} :: 60000 : X = 108{,}620$ cub. ft. per minute.

Example on Q13.—The coefficient of friction is 0·004 in. W.G. per 1000 ft., and the quantity of air passing per minute is 200,000 cub. ft. By cleaning and arching the friction is reduced to 0·0024 in.; what is now the quantity passing ?

$$\frac{1}{\sqrt{0{\cdot}004}} : \frac{1}{\sqrt{0{\cdot}0024}} :: 200000 : X = 258{,}100 \text{ cub. ft. per minute.}$$

General Example.—An airway 8 ft. by 5 ft. passes 30,000 cub. ft. per minute, how much air will it pass if the section be increased to 12 ft. by 6 ft., and the power and coefficient of friction remains the same ?

h.p. $\frac{k_1 s_1 Q_1^3}{33000 A^3} = \frac{k_2 s_2 Q_2^3}{33000 A_2^3} \therefore \frac{26 \times 30000^3}{40^3 \times 33000} = \frac{36 \times Q_2^3}{72^3 \times 33000} = 48{,}440$ cub. ft. per min.

THE LAWS OF PRESSURE.

p1. By (171) pressure varies DIRECTLY as the square of the velocity, *i.e.* as v^2.
p2. ,, (175) ,, ,, ,, square of the quantity, *i.e.* as Q^2.
p3. ,, (172) ,, ,, ,, coefficient of friction, *i.e.* as k.
p4. ,, (173) ,, ,, ,, rubbing surface (velocity constant), *i.e.* as s.
p5. ,, (106) ,, ,, ,, square of the fan speed, *i.e.* as V^2.
p6. ,, (106) ,, ,, ,, manometrical efficiency, *i.e.* as K.

THE LAWS OF PRESSURE—*continued.*

p7. By (176) pressure varies DIRECTLY as the square of the cube root of the power in the air (for a constant quantity), *i.e.* as $u^{\frac{2}{3}}$.

p8. ,, (26) ,, ,, ,, reciprocal of the quantity (for a constant power in the air), *i.e.* as $\frac{1}{Q}$.

p9. ,, the Prussian Firedamp Committee the pressure varies as the 2/3 power of the density of the air, *i.e.* as $d^{\frac{2}{3}}$.

p10. ,, (117) pressure varies INVERSELY as the square of the mine orifice or as $\frac{1}{a^2}$.

p11. ,, (170) ,, ,, ,, area of the airway, or as $\frac{1}{a}$.

p12. ,, (179) ,, ,, ,, the fifth power of the airway diameter or as $\frac{1}{D^5}$.

*p*4, *p*11, and *p*1 above are known as the first, second, and third laws of mine-friction respectively.

The Application of Laws of Pressure.

*Example on p*1 *or p*2.—400,000 cub. ft. of air pass in a mine when the W.G. is 4 in. What is the effect on the pressure if the volume is allowed to fall to half the above amount ?

$$4^2 : 2^2 :: 4 : X = 1 \text{ in. water gauge.}$$

*Example on p*3.—If a head of 120 ft. of air column is needed to overcome a mine resistance the coefficient of which is 10^{-8}, what head will be required if the friction be as stated by Atkinson ?

$$10^{-8} : 0{\cdot}0217 \times 10^{-6} :: 120 : X = 260{\cdot}4 \text{ ft. head.}$$

(For Example on *p*4, see Example on *p*11.)

*Example on p*5.—A ventilator running at 160 r.p.m. and producing $2\frac{1}{2}$ in. W.G. is speeded up 25 per cent. Find the effect on the water gauge.

$$160^2 : 200^2 :: 2{\cdot}5 : X = 3{\cdot}9 \text{ in. W.G.}$$

*Example on p*6.—The ventilating pressure is 16 lbs. per square foot previous to the manometrical efficiency falling to $\frac{3}{4}$ths its normal value. What is the resulting ventilating pressure ?

$$\tfrac{3}{4} \times 16 = 12 \text{ lbs. per sq. ft.}$$

*Example on p*7.—When the W.G. is 2 in. the power is 50, what will the gauge read if the power is increased to 60 without affecting the circulation ?

$$\left(\frac{60}{50}\right)^{\frac{3}{2}} \times 2 = 2{\cdot}258 \text{ in. W.G.}$$

*Example on p*8.—The W.G. at a certain mine is 4 in. when the quantity is 200,000 cub. ft. per minute, and without affecting the power in the air this volume is raised to 250,000 by dimensional alterations ; state the new pressure required.

$$\frac{1}{200} : \frac{1}{250} :: 4 : X = 3{\cdot}2 \text{ W.G.}$$

*Example on p*9.—On a certain day in winter the barometer stands at 30 in. and the thermometer reads 41° F. These instruments read 29½ in. and 85° F. respectively on a certain summer day. If in the former case the ventilating pressure was 2·75 in. W.G., what would it be on the latter occasion ?

$$\left(\frac{1{\cdot}3212 \times 30}{500}\right)^{\frac{3}{2}} : \left(\frac{1{\cdot}3213 \times 29{\cdot}5}{544}\right)^{\frac{3}{2}} :: 2{\cdot}75 : X = 2\tfrac{1}{2} \text{ in. W.G.}$$

Example on p. 10.—The equivalent orifice of a mine showing 4 in. W.G. is 18 sq. ft. ; what would the pressure be if the orifice be considered 36 sq. ft.?

$$\frac{1}{(18)^2} : \frac{1}{(36)^2} :: 4 : X = 1 \text{ in. W.G.}$$

*Example on p*11 and *p*4.—The water gauge reads 2 in. in a 6 ft. square airway ; what will be the reading when the airway is enlarged to 9 ft. square ?

$$\frac{24}{36} : \frac{36}{81} :: 2 : X = 1{\cdot}33 \text{ in. W.G.}$$

*Example on p*12.—A circular shaft which absorbs 1½ in. of water gauge is being widened from 14 ft. to 21 ft. ; what pressure will it take when completed if the coefficient of friction is constant ?

$$\frac{1}{(14)^5} : \frac{1}{(21)^5} :: 1\tfrac{1}{2} : X = 0{\cdot}1975 \text{ in. W.G.}$$

Before discussing the laws of power it will be well to observe that it is power which produces ventilation and not pressure—the latter is the product or result of ventilation.

By considering together (170) and (176), as in *p*7 above, it

will be seen that for a constant circulation the cube root of the power varies as the square root of the pressure, but for a constant power in the air $u - pQ$ as given by (26), see *p*8. It will also be clear that most of the preceding ratios are capable of reciprocal and joint expression in many forms, and that these ratio types will multiply with the number of conditions assumed constant. Thus by taking the portion under the root sign in the split formula (201) as being constant, it can be shown that Q^5 varies as n^4, and consequently $(W.G.)^5$ varies as n^8, where n is the number of splits. Again, if the fan-shaft power be assumed constant, Q will vary as the $\frac{4}{5}$ths power of $\frac{a}{\sqrt[3]{s}}$, whilst if the power in the air be taken as constant instead, this fractional index is replaced by unity, and the pressure will vary inversely as the quantity as in *p*8. The combinations and permutations of these terms are far too extensive to treat exhaustively here, therefore we will restrict ourselves to the more general descriptions of the laws of mine ventilation and leave the possible allied forms to the reader's ingenuity. The basis of our laws will be, of course, Atkinson's formula.

THE LAWS OF POWER.

u1.	**By (178)**	**power varies**	**DIRECTLY**	**as the**	**cube of the quantity, *i.e.* as Q^3.**
u2.	**,, (178 and (154)**	**,,**	**,,**	**,,**	**cube of the fan speed, *i.e.* as V^3.**
u3.	**,, (176)**	**,,**	**,,**	**,,**	**coefficient of friction, *i.e.* as *k*.**
u4.	**,, (176)**	**,,**	**,,**	**,,**	**rubbing surface, *i.e.* as *s*.**
u5.	**,, (152) and (178)**	**,,**	**,,**	**,,**	**square root of the manometrical efficiency cubed *i.e.* as $K^{\frac{3}{2}}$.**
u6.	**,, (175) and (178)**	**,,**	**,,**	**,,**	**square root of the pressure cubed, *i.e.* as $p^{\frac{3}{2}}$.**

The Application of the Laws of Power.

*Example on u*1.—A volume of 200,000 cub. ft. per minute is provided when the power is 50 ; what power will be needed to double the quantity ?

$$200^3 : 400^3 :: 60 : X = 480.$$

Example on u2.—The speed of a fan developing 40 h.p. is increased from 150 to 200 r.p.m.; what is the horse-power required to maintain this speed ?

$$150^3 : 200^3 :: 40 : X = 94{\cdot}8 \text{ h.p.}$$

Example on u3.—When the coefficient of friction is 0·02 lb. per 1000 ft., the power expended in a certain airway is 15 h.p., what will the power be if the friction be halved ?

$$0{\cdot}02 : 0{\cdot}01 :: 15 : X = 7\tfrac{1}{2} \text{ h.p.}$$

Example on u4.—An airway 6 ft. by 7 ft. and 2000 ft. long is being remade to 8 ft. by 8 ft.: if the power expended in ventilating it was formerly 14 h.p., what will it be when completed if the velocity is kept the same ?

$$26 : 32 :: 14 : X = 17{\cdot}2 \text{ h.p.}$$

Example on u5.—On reorganising the ajutages of a certain fan the manometrical efficiency is increased from 0·65 to 0·7. If the old power "in the air" was 100 h.p., what will it be now ?

$$0{\cdot}65^{\frac{2}{3}} : 0{\cdot}7^{\frac{2}{3}} :: 100 : X = 111{\cdot}8 \text{ h.p.}$$

Example on u6.—To what extent must the power be increased in order to double the pressure ?

$$\frac{p_2}{p_1} = 2 \text{ and } u^{\frac{1}{3}}_1 : u^{\frac{1}{3}}_2 :: p^{\frac{1}{3}}_1 : p^{\frac{1}{3}}_2$$

$$\therefore \sqrt[3]{\frac{u_1}{u_2}} \quad \sqrt{\frac{p_1}{p_2}} = \sqrt{2^3} = \sqrt{8} = 2{\cdot}828 \text{ times.}$$

THE LAWS OF VELOCITY.

v1. By (174) velocity varies DIRECTLY as the square root of the pressure or area, *i.e.* as $\sqrt{p}$ or $\sqrt{a}$.

v2. ,, (178) ,, ,, ,, cube root of the power, *i.e.* as u^3.

v3. ,, (174) ,, INVERSELY ,, square root of the coefficient of friction or the rubbing surface, *i.e.* as $\frac{1}{\sqrt{k}}$ or $\frac{1}{\sqrt{s}}$.

The Application of the Laws of Velocity.

Example on v1.—When the pressure is increased by 50 per cent., what is the effect on the velocity ?

$1^{\frac{1}{2}} : 1{\cdot}5^{\frac{1}{2}} :: v_1 : v_2 =$ the velocity is increased 1·22 times.

*Example on v*2.—45 h.p. maintains a mine velocity of 1100 ft. per min.; what will the velocity be if the power is doubled?

$$45^{\frac{1}{3}} : 90^{\frac{1}{3}} :: 1100 : X = 1395 \text{ ft. per minute.}$$

*Example on v*3.—In an airway 7 ft. × 9½ ft. × 4000 ft. a velocity of 900 ft. per minute is maintained against a resistance the coefficient of which is 0·015 × 10^{-6}. State the velocity if (i) the length is halved; (ii) if Atkinson's coefficient is adopted?

(i) Here p and a are constant

$$\therefore \frac{1}{\sqrt{4000}} : \frac{1}{\sqrt{2000}} :: 900 : X = 1272{\cdot}6 \text{ ft. per minute}$$

(ii) Here 10^{-8} is common

$$\therefore \frac{1}{\sqrt{1{\cdot}5}} : \frac{1}{\sqrt{2{\cdot}17}} :: 900 : X = 748 \text{ ft. per minute}$$

In a similar manner the laws respecting the coefficient of friction, area, rubbing surface, etc., can be conveniently stated, but in order to avoid repetition the matter will now be left to the reader, and the remainder of this chapter devoted to various problems.

Problems on Fans.

Example 1.—The peripheral speed of a simple fan is 100 ft. per second; if the density of the air is 0·076 lb. per cub. ft., find the theoretical water gauge.

By (102) $$\frac{100^2 \times 0{\cdot}076}{32{\cdot}2 \times 5{\cdot}2} = 4{\cdot}538 \text{ in. theoretical W.G.}$$

Example 2.—In the foregoing example the observed water gauge in the fan drift is 3 in., what is the manometrical efficiency?

$$\frac{\text{Observed W.G.}}{\text{Theoretical W.G.}} = \frac{3}{4{\cdot}538} \times 100 = 66 \text{ per cent. manometrical efficiency.}$$

Example 3.—A fan 150 in. in diameter is driven by ropes with a speed ratio of 1 : 3, and the engine is a double-acting steam-engine which makes 70 r.p.m. What is the peripheral velocity of the fan tips?

$$\frac{3 \times 70 \times 12{\cdot}5 \times \pi}{60} = 137{\cdot}4 \text{ ft. per second.}$$

Example 4.—If the above fan deals with 200,000 cub. ft. per minute, and the air is of a density of 0·075 lb. per cub. ft., what is the horse-power transmitted to the air ?

By (36) $$\frac{200000 \times 0{\cdot}075 \times 137{\cdot}4 \times 137{\cdot}4}{60 \times 550 \times 32{\cdot}2} = 266{\cdot}5 \text{ h.p.}$$

Example 5.—If the fan-drift water gauge shows 4·5 in. in the above case, what is the manometrical or true mechanical efficiency ?

By (136) $$\frac{5{\cdot}2 \times 4{\cdot}5 \times 32{\cdot}2 \times 100}{(137{\cdot}4)^2 \times 0{\cdot}075} = 53{\cdot}2 \text{ per cent.}$$

Example 6.—A fan which circulates 150,000 cub. ft. of air per minute against 2 in. W.G., is driven by a direct-current motor taking 500 volts at 120 amps. and 95 per cent. efficient ; what is the fan efficiency ?

$$\frac{500 \times 120 \times 0{\cdot}95}{746} = 76{\cdot}4 \text{ h.p. supplied to the fan.}$$

$$\frac{2 \times 5{\cdot}2 \times 150000}{33000} = 47{\cdot}28 \text{ h.p. in the air.}$$

$$\frac{47{\cdot}28}{76{\cdot}4} = 61{\cdot}87 \text{ per cent.}$$

Example 7.—A fan delivers air at a tip speed of 130 ft. per second and maintains a water gauge of 5 in. ; if the total efficiency of the whole plant is 64 per cent., what is the engine-drive efficiency ? (Air here weighs 0·075 lbs. per cub. ft.)

By (136)

$$\frac{5{\cdot}2 \times 5 \times 32{\cdot}2}{130 \times 130 \times 0{\cdot}075} = 66 \text{ per cent. manometrical efficiency.}$$

By (141)

$$\frac{64}{66} \times 100 = 97{\cdot}0 \text{ per cent. drive efficiency or } 3{\cdot}0 \text{ per cent. friction,}$$

etc., losses in the drive.

Example 8.—If a fan be considered theoretically perfect, what will be its initial water gauge if it runs at 120 r.p.m. when the inlet is closed ? The diameter is $12\frac{1}{2}$ ft.

By (104)

$$0{\cdot}00045V^2 = 0{\cdot}0045D^2N^2 = 0{\cdot}0045\left(\frac{12\frac{1}{2} \times 120}{60}\right)^2 = 2{\cdot}8 \text{ in. initial W.G.}$$

Example 9.—A fan with an evasée 7 ft. square at the top produces 100,000 cub. ft. per minute; if the chimney is extended until the top area is 12 ft. square what will be the gain in pressure (take the air as weighing 0·075 lb. per cubic foot)?

Velocity when top is 7 ft. × 7 ft. $= \frac{100000}{49 \times 60} = 34$ ft. per second.

,, ,, ,, 12 ft. × 12 ft. $\times \frac{100000}{144 \times 60} = 11{\cdot}57$ ft. per second.

Energy lost in the first case is, by (10) $= \frac{100000 \times 0{\cdot}075 \times 34^2}{60 \times 2 \times 32{\cdot}2} = 2243{\cdot}8$ ft.-lbs.

,, ,, second ,, ,, $= \frac{100000 \times 0{\cdot}075 = 11{\cdot}57^2}{60 \times 2 \times 32{\cdot}2} = 259{\cdot}8$ ft.-lbs.

Energy reclaimed by new chimney $= 2243{\cdot}8 - 259{\cdot}8 = 1984$ ft.-lbs.

Corresponding pressure $= \frac{\text{ft.-lbs.}}{\text{lbs.}} = \frac{1984 \times 60}{100000 \times 0{\cdot}075} = 15{\cdot}87$ feet head.

By (93) ,, water gauge $= \frac{15{\cdot}87 \times 0{\cdot}075 \times 12}{62{\cdot}3} = 0{\cdot}23$-in. W.G.

Example 10.—A double inlet Sirocco fan has to pass 200,000 cub. ft. of air per minute against a $2\frac{1}{2}$ in. W.G.; find its inlet diameter.

By (158) fan diameter $= \frac{0{\cdot}354 \times \sqrt{200000}}{\sqrt[4]{2{\cdot}5}} = 126$ in. diam.

Assuming that the fan inlet is $87\frac{1}{2}$ per cent. of the diameter,

inlet diameter $= \frac{87\frac{1}{2}}{100} \times 126 = 110\frac{1}{4}$ in., say 9 ft. 3 in.

Example 11.—Two fans working separately produce quantities Q_1 and Q_2 when the pressures are p_1 and p_2 respectively: when they work together in series on the mine the quantity Q_3 is yielded with a pressure of p_3; find this total quantity.

$$\begin{aligned} &\text{First Power} &&= u_1 = p_1 Q_1. \\ &\text{Second Power} &&= u_2 = p_2 Q_2. \\ &\therefore \quad u_1 + u_2 &&= p_1 Q_1 + p_2 Q_2 = p_3 Q_3. \end{aligned}$$

Mine resistance being constant.

$$\frac{Q^2{}_3}{p_3} = \frac{Q^2{}_2}{p_2} + \frac{Q^2{}_1}{p_1}$$

$$\therefore \frac{Q^2{}_3}{p_3} \times p_3 Q_3 = \left(\frac{Q^2{}_2}{p_2} \times p_2 Q_2\right) + \left(\frac{Q^2{}_1}{p_1} \times p_1 Q_1\right)$$

$$\therefore Q_3 = \sqrt[3]{Q^3{}_2 + Q^3{}_2} \quad \ldots \quad (220)$$

Problems on Apportionment.

Example 1.—Between two points in a rectangular airway 8 ft. $\times 7\frac{1}{2}$ ft. and 1500 ft. long, a $\frac{1}{2}$ in. W.G. is maintained and also a velocity of 500 ft. per minute. What amount of air would pass if another exactly similar airway were added between the two points and the same pressure maintained in each ?

Pressure being the same for each split as for the single airway, and all other things being equal, the same quantity must pass in each as in the single air, and so the total quantity now passing is—

$$2 \times 500 \times 7\tfrac{1}{2} \times 8 = 60{,}000 \text{ cub. ft. per minute.}$$

Example 2.—In two mines the cross-sectional dimensions of the airways are similar, and in one of them the total length of the airway is three times that of the other; what is the relation between the equivalent orifice of the first or short mine and that of the second or long mine ?

It will be seen here that the pressures are proportional to the lengths, and by (117) the equivalent orifices are inversely proportional to the square root of the water gauges or ventilating pressures. Thus the equivalent orifices are in the proportion $\frac{\sqrt{3}}{\sqrt{1}}$. Therefore in the short one the orifice is 1·732 times that of the long mine.

Example 3.—40,000 cub. ft. of air pass per minute through a single airway: what quantity will pass if there are two such airways similar to each other ?

(i) If the power remains constant ?
(ii) " " pressure " " ?

(i) Being similar, half the required quantity will pass through each split, and half the initial h.p. will be expended on each; but horse-power is, by *a*1 or (178), proportional to the cube of the quantity, therefore

$$\frac{\text{h.p.}_1}{\text{h.p.}_2} = \frac{Q^3{}_1}{Q^3{}_2} \text{ or } \frac{1}{0{\cdot}5} = \frac{40000^3}{Q^3{}_2}.$$

$\therefore Q_2 = \sqrt[3]{40000^3 \times 0{\cdot}5} = 31{,}748$ cub. ft. per minute in each airway. Total quantity $= 2 \times 31748 = 63{,}496$ cub. ft.

(ii) The pressure in this case being constant, the quantity passing will be in proportion to the root of the number of splits.

$$\therefore\ 40000 \times \sqrt{2} = 56{,}560 \text{ cub. ft.}$$

Example 4.—What quantity passes through a regulator $2\frac{1}{4}$ ft. square with a water gauge of $\frac{3}{4}$ in. (take the weight of air as 0·08 lbs.) ?

By (117 *et seq.*) $Q = 60 \times 0{\cdot}65 \times 2\frac{1}{4}^2 \times \sqrt{\frac{62{\cdot}3 \times 0{\cdot}75}{12 \times 0{\cdot}08} \times 2g} = 11{,}054$ cub. ft. per min.

Example 5.—A water gauge shows a difference of pressure between the opposite sides of a regulator to be 1 in. The area of the opening is 3 sq. ft., what would be the average volume and velocity of air passing per minute supposing the opening to be round ?

By (117) $Q = \frac{a\sqrt{\text{W.G.}}}{0{\cdot}388} = \frac{3 \times \sqrt{1}}{0{\cdot}388} = 7{\cdot}731$ thousands $= 7731$ cub. ft. per min.

,, (117 *et seq.*) $v = \frac{Q}{0{\cdot}65a} = \frac{7731}{0{\cdot}65 \times 3} = 3965$ ft. per min.

Example 6.—40,000 cub. ft. pass per minute along an airway $7\frac{1}{2}$ ft. × 8 ft. and 2000 ft. long; what effect would a regulator of 8 sq. ft. aperture have upon the quantity passing ?
By (175) pressure required to pass the quantity before regulating

$$p = \frac{0{\cdot}01 \times 62{,}000 \times 40 \times 40}{60 \times 60 \times 60} = 4{\cdot}6 \text{ lbs. per sq. ft.}$$

By (117) resistance due to regulator =

$$\left(\frac{0{\cdot}388 \times Q}{8}\right)^2 \times 5{\cdot}2 = \left(\frac{0{\cdot}388 \times 40}{8}\right)^2 \times 5{\cdot}2 = 19{\cdot}57 \text{ lbs. per sq. ft.}$$

Required total pressure to pass the quantity = 4·6 + 19·57 = 24·17 lbs. per sq. ft. Without the regulator the power needed to pass the quantity is, by (22) =

$$\frac{40000 \times 4{\cdot}6}{33000} = 5{\cdot}57 \text{ h.p.}$$

$$\text{With the regulator power needed} = \frac{40000 \times 24{\cdot}17}{33000} = 29{\cdot}31 \text{ h.p.}$$

Quantities here vary as the cube roots of the powers :—

$\sqrt[3]{29{\cdot}30} : \sqrt[3]{5{\cdot}57} :: 40000 : X = 23{,}000$ cub. ft. passes per min. on regulating.

Example 7.—A mine is ventilated by two splits—(*a*) 10,000 ft. long and (*b*) 6000 ft. long., and each of the same cross-sectional

dimensions, 8 ft. by 8 ft. The ventilating pressure is 20 lbs. per sq. ft. It is required that the amount of air passing into the longer split should be twice that passing into the shorter one. Find the quantity of air for each split and the size of regulator opening required to maintain such a proportion.

The natural quantity for each split will be—

For split (*a*) by (175)

$$Q = \sqrt{\frac{pa^3}{ks}} = \sqrt{\frac{20 \times 64^3}{0{\cdot}01 \times 32 \times 10000}} = 40{,}500 \text{ cub. ft. per minute.}$$

For split (*b*) by (175)

$$Q = \sqrt{\frac{pa^3}{ks}} = \sqrt{\frac{20 \times 64^3}{0{\cdot}01 \times 32 \times 6000}} = 52{,}256 \text{ cub. ft. per minute.}$$

This latter quantity must be reduced by the terms of the question to half that in (*a*), or 20,250 cub. ft. per minute.

The mine water gauge is $\frac{20}{5{\cdot}2} = 3{\cdot}85$ in., and that required to pass half of the quantity of (*a*) is (by rule *p*2) as follows :—

$$3{\cdot}85 : X :: 52256^2 : 20250^2 = 0{\cdot}58 \text{ in.}$$

Therefore the W.G. that must be absorbed is $3{\cdot}85 - 0{\cdot}58 = 3{\cdot}27$ in.

The area of a regulator to pass the amount stated at the above W.G. is by (117)

$$a = \frac{0{\cdot}388 \times Q}{\sqrt{\text{W.G.}}} = \frac{0{\cdot}388 \times 20{\cdot}25}{\sqrt{3{\cdot}27}} = 4{\cdot}34 \text{ sq. ft. area of regulator orifice.}$$

Example 8.—Abandoned workings are found to contain an inflammable mixture of firedamp and air, there being 6·4 per cent. of firedamp present. If the mixture occupies 50,000 cub. ft., how much air will be required to dilute it down to $1\frac{1}{4}$ per cent. of gas ?

Let Q be the quantity of gaseous mixture in cub. ft.
„ Q_1 „ „ „ „ additional air required for dilution.
„ A „ „ percentage of gas in the mixture before dilution.
„ A_1 „ „ „ „ „ „ after „

Then $Q_1 = Q \times \left(\frac{A - A_1}{A_1}\right)$.

$$\therefore\ Q_1 = \frac{50000 \times (6{\cdot}4 - 1{\cdot}25)}{1{\cdot}25} = 206{,}000 \text{ cub. ft. of air.}$$

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Dictionary of Applied Physics, Glazebrook; *Min. and Eng. Journal Press*, 1922; *Mining*, Arnold Lupton; *Natural Philosophy of a Ventilating Regulator*, Halbaum; *Mine Gases and Ventilation*, Beard: *Hydro-Electrical Engineering*, vol. i., Gibson.

CHAPTER VIII

THE PRACTICE OF MINE VENTILATION

As in most other industries, definite symbols are used in the practice of mine ventilation in order to depict some more or less difficult practical feature upon the plan. The General Regulations of 1919 recommend, in the First Schedule, the code of ventilation signs shown in fig. 131, and the C.M.A., 1911, sections 20 (2), 41, and 42, make further stipulations as to the ventilating accessories and plans; among these we find those for stoppings.

A Stopping.—This, as the name implies, is an obstruction built at the entrance to a district to seal off a fire or water, or it may be built to separate intake and return. Those separating main intake and main return are required by sec. 42 (3) of the Act to be so constructed as not to be liable to destruction in case of an explosion. General Regulation 91 provides the manner of the construction to be 5 yards packing, or 3 yards packing with 9-in. mortar-faced brick or concrete, the space between the roadway and the face of the stopping to be kept clear.

Brattice Sheeting.—This is prepared canvas treated with carbonaceous bye-products and usually oiled on one face, which face is generally on the windward side. It is kept in rolls and is probably the most serviceable accessory in ventilating practice. It should be nailed with short, broad-headed nails on the windward side of the bar or collar, and in the case of a heavy sheet, should be doubled back at the top and nailed to a securely screwed-in lath $\frac{3}{4}$ in. thick, the vertical folds overlapping about 18 in. and nailed to the vertical posts, as in fig. 133. Sheets are shown on plans as either sinuous lines or short straight lines across the airways. Figs. 91–93 show the bratticing of an entry, and in such cases the intake should be the narrower side so as to accommodate the increased volume due to heating and gases at the face, which volume will now pass out through the broader return part.

Bratticing in sinking and other shafts is no longer popular owing to the damage done by "nicking" the temporary wooden cross buntons and sheathing into the brickwork.

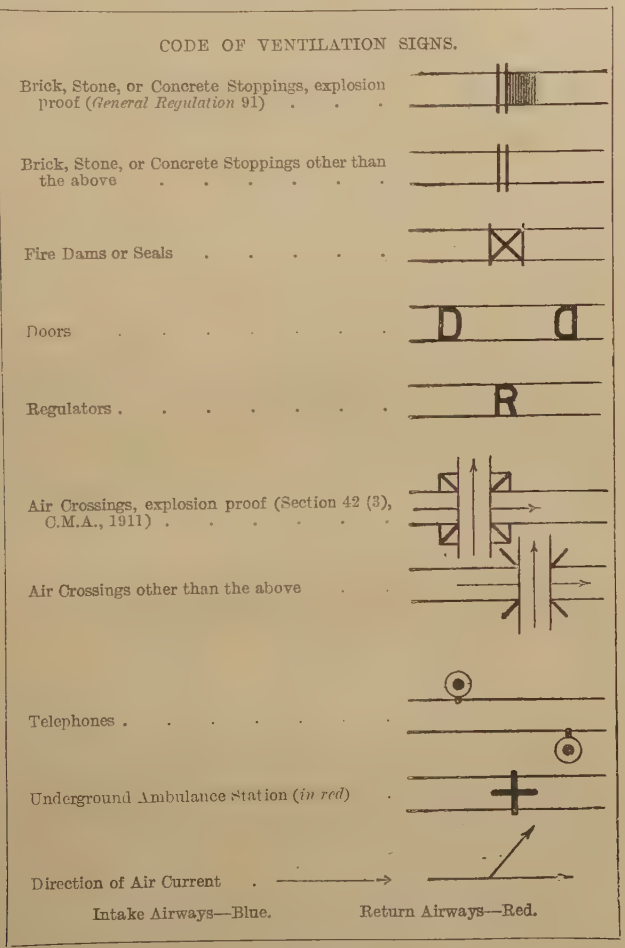

Fig. 131.

A complete sheet is known as a block sheet, a half sheet (horizontally) as a hurdle, and a half-hung one as a skeleton sheet.

Doors.—These may be of wood or sheet steel, and should be hinged on the windward side as in fig. 134. They should lean

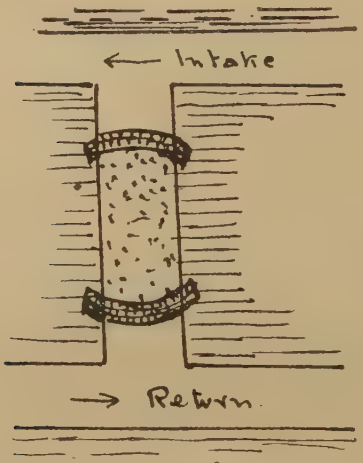

FIG. 132.—Stopping.

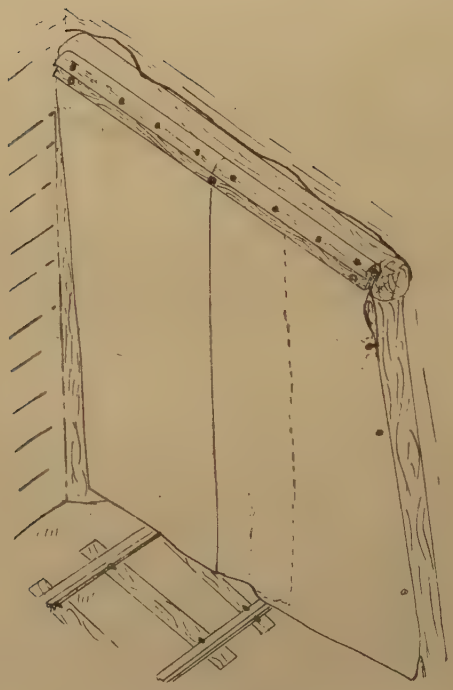

FIG. 133.—Brattice.

over in the head, from the direction of the pressure, to enable them to close by gravity : skewed hinges are not to be recommended for this purpose as they are difficult to keep airtight in bad ground. Doors are usually sealed against the strata by concrete on brickwork, by packs, or by turf. Turf is very extensively used in some places, especially for sealing roof cavities, to avoid awkward sheeting. It has the advantage of considerable yield under " weighting " and also airtightness. Fig. 136 shows an automatic or self-acting door as used on mechanical haulage planes. The lever arrangement will need no further explanation.

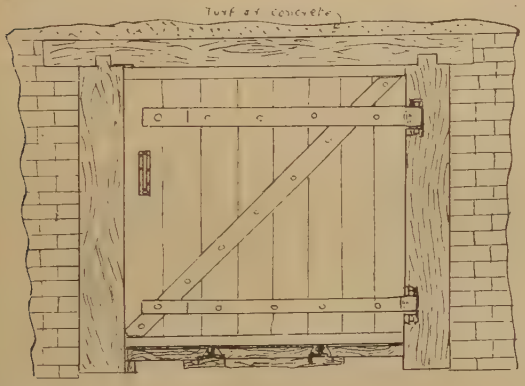

Fig. 134.

Fig. 135.

We will not dwell long on automatic doors on main planes, because in such positions they are decidedly bad mining practice and should not be introduced on busy sections if they can possibly be avoided. They are not always so automatic as popularly supposed, and need very careful supervision and frequent repairs. Their best use is upon surface air locks, and in some cases main separation roads.

Notes on Mine Doors.—(*a*) In mines liable to fire, the doors and frames should be fireproofed in sheet iron, well built, and well supervised.

(*b*) High-pressure doors should be latched, because in an explosion these doors are often the pivotal point of the " pioneer wave."

(*c*) Every district or isolated split should have a system of open doors at its entry and exit to assist in isolating it in case of fire.

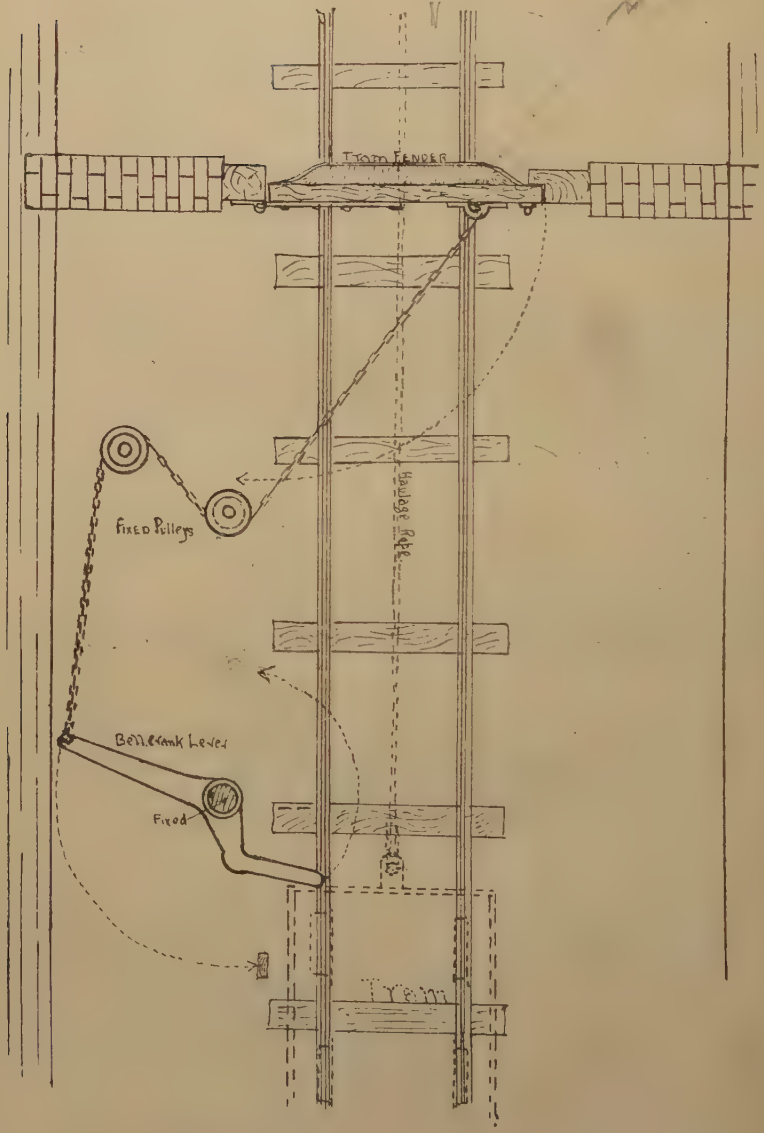

FIG. 136.—Self-acting door.

(*d*) All doors, especially those in (*c*) above, should close flush with the jamb, and not in a groove or tenon or other recess likely to grip the door in case of fire and so prevent it opening.

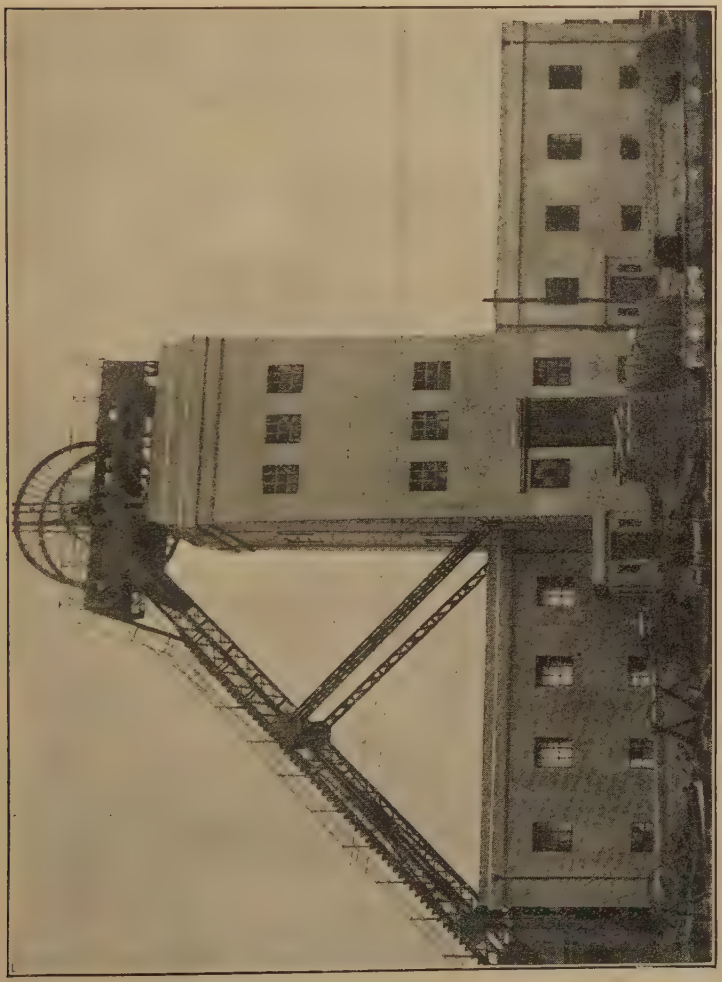

FIG. 137.—External view of airlock at Askern Main in reinforced concrete. (By Industrial Constructions, Ltd.)

(*e*) All door systems on busy sections should be double, treble, or of the airlock type.

(*f*) Emergency doors should have a notice attached stating their purpose and operation in case of fire, reversal, etc.

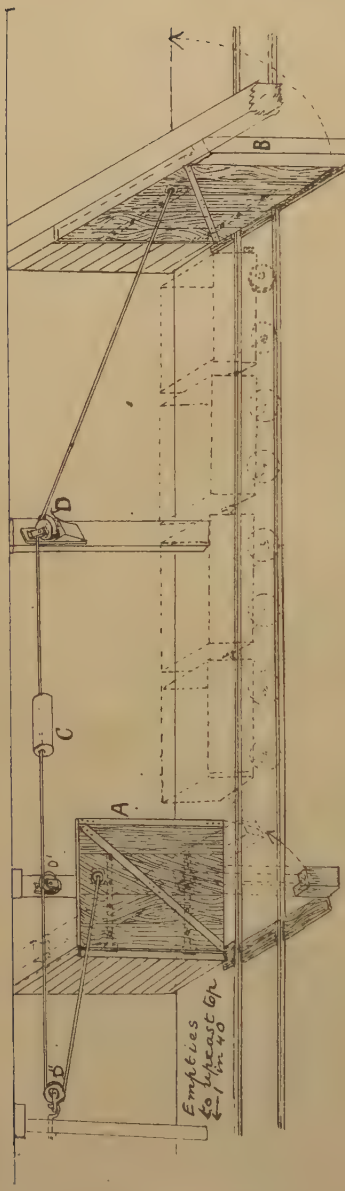

FIG. 138.—Four tram airlock (Ramsay).

(*g*) A notice of the "Safety First" type, *i.e.* SHUT THIS DOOR, should be attached to every working door in the mine.

Airlocks.—These are very valuable adjuncts to the ventilating system, because they economise air. They are used on cased upcasts (fig. 137) where cage operated doors or bonnets are not desirable as in the case of regular and intensive coal winding. Separation doors below ground are also very suitable places for airlock devices (see fig. 138). The airlock itself consists essentially of two doors, A and B, the one to be kept closed if the other is open; or both closed together. The arrangement shown in fig. 138 is very simple. It will be noticed that only one door at a time can be opened, and that only when the other is closed. The balance weight, C, is employed simply to overcome the friction of the snatch block and pulleys D D′, etc. Both doors operate in the same direction, *i.e.* they close, like all other mine doors, in the direction of the action of the greater air pressure. In other cases it is usual to have an electric signal light worked by a bell crank contact-maker off door A, which completes the

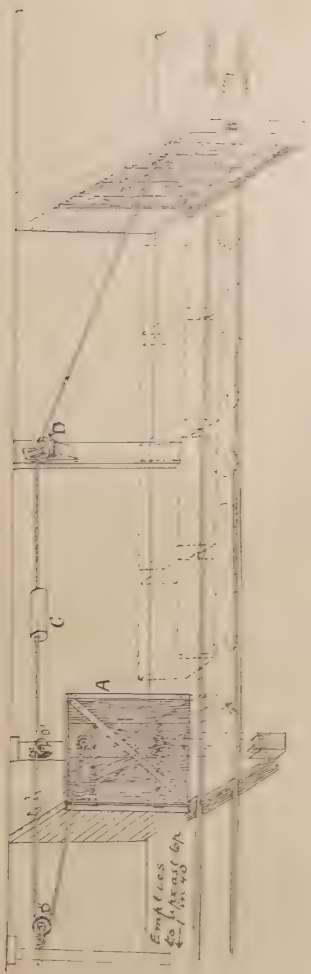

FIG. 138.—Four tram airlock (Ramsay).

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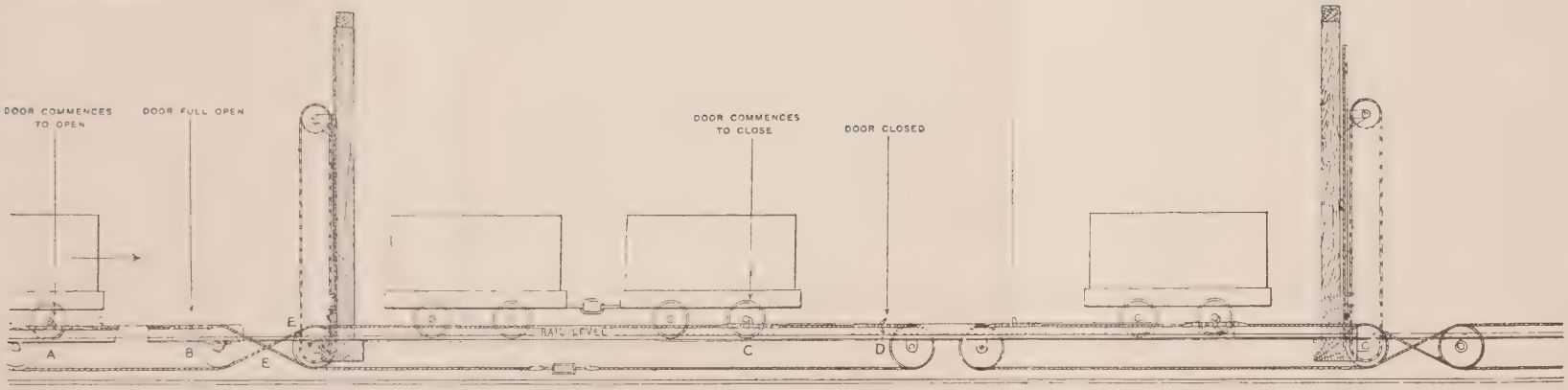


FIG. 139.

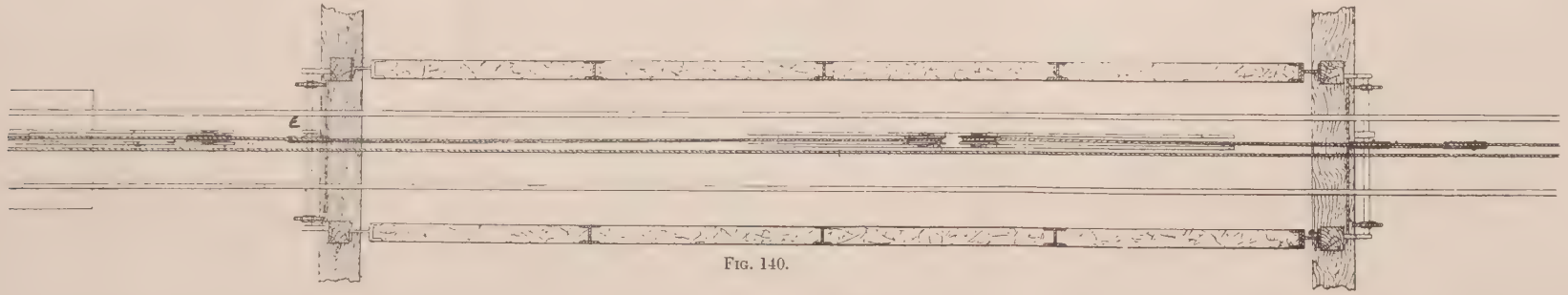


FIG. 140.

FIG. 141.

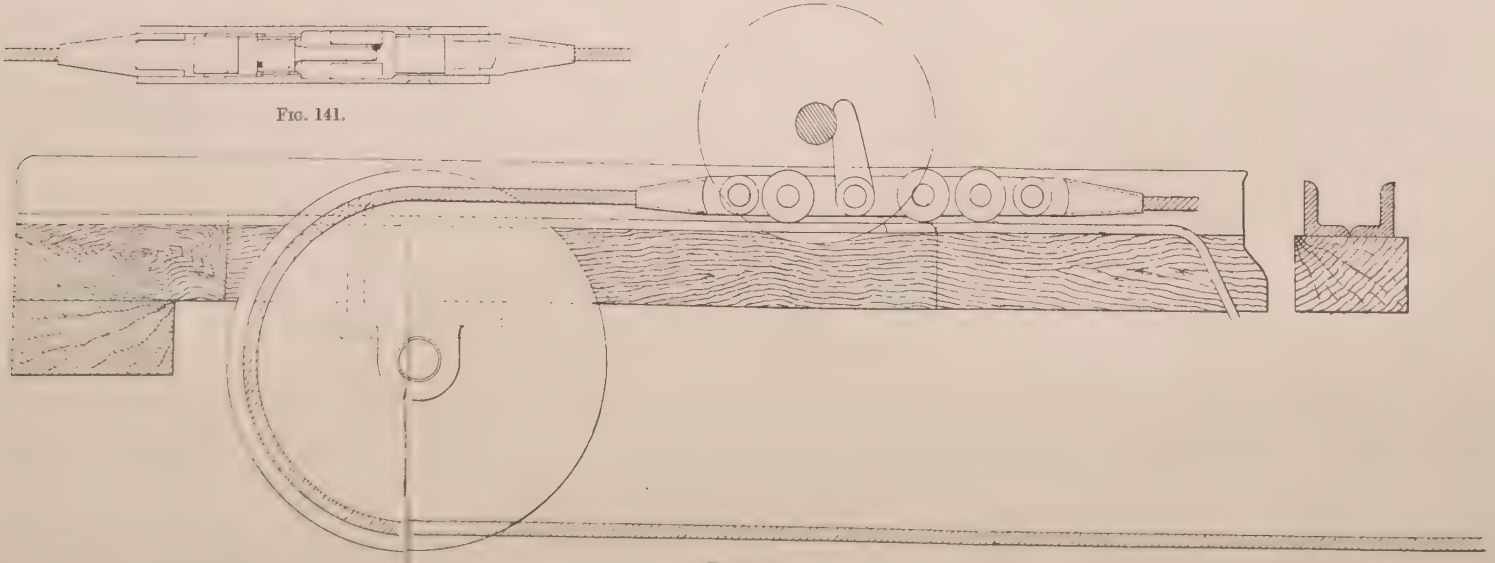


FIG. 142.
Automatic airlock doors.

signal circuit when the door opens, thus illuminating a red globe on the outside of B, to warn the workmen that B must not be opened: otherwise a window may be fitted for sighting through. The idea of axle kick levers will be easily applicable also. The above airlock door motion (fig. 138), if applied to vertical sliding doors, needs a much more expensive roof or floor mechanism, so that where space considerations precludes the installation of swing doors, bobweight balanced vertical sliders operated by a boy are usually installed instead.

Fig. 145 shows a method of mechanically operating the airlock doors of a double-deck aircasing. The top-deck door, which is the smaller, opens outward, and the pressure is just sufficient to open it: but the greater area of the bottom door, working on the same vertical shaft, keeps it closed. The lower door opens inward, and by the above arrangement needs very little expenditure of power. The engine may be controlled by a hand lever or by a simple kick-over crank lever engaging with the front tub wheel-axle on the inner side and another on the outer side of the doors for closing. Closure may also be brought about by balance bobs and cut-offs, by gravity, or by mechanical means.

A suitable mechanical arrangement which has been well tried is that described in a paper by Mr Clement Fletcher before the Institution of Mining Engineers, 1918–19, the essentials of which are as follows:—

The automatic doors (fig. 139) are made of sheet iron $\frac{1}{8}$ in. thick, 4 ft. 3 in. high by 3 ft. 10 in. wide, braced with **T**-iron, and working vertically in a machined gun-metal groove on two Hans Renolds chains. The door is attached to each end of the chain; it passes over a counter shaft, round the sprocket wheels on the driving shaft E (figs. 139 and 140), back to the door, so forming an endless chain. The driving shaft, which is below the tub rails, carries, in addition to the sprockets, a driving wheel 12 in. diam. keyed on. Around the wheel a $\frac{3}{8}$-in. wire rope passes twice, then round return wheels, and is attached to a tripper or operating catch. This catch is in the form of an **L** leg and is mounted in a small travelling carriage running in a channel-iron groove fixed 3 in. clear of the haulage rope and acting as follows. When the tub clipped to the haulage rope arrives at A, fig. 139, the axle engages the clip and moves it along, so revolving the driving shaft and sprockets and opening the door. When the door is fully open the tub has travelled to B; at this point it is automatically released from the axle, the tub passing on through the doors to C,

where the axle again engages another catch, which reverses the driving shaft and closes the door by the time the tub gets to D. The tubs pass on to a similar arrangement for operating the other door or doors. The disengagement of the operating catch from the axle is brought about by means of a groove in the channel iron in which it runs; at a fixed point the horizontal limb of the **L** leg drops through a cut-away part of the groove and so frees the vertical limb of it from the axle and, the rope being practically endless, it arrives at the action point at the end of the reversal run.

The idea, though designed for separation doors, can be easily adapted to any other kind of airlock, and any number of doors at any desired distance with or without a haulage rope for tub clipping, *i.e.* by gravity.

The airlock casing itself is usually made of light steel sections and brickwork, or in ferro-concrete as at Bentley. Figs. 143 to 145 show the details of the airlocks at Maltby Main Colliery. It is built throughout of brickwork walls, with roofing and flooring of light rolled steel joists covered by wooden sleepers, and these in turn with cement and a neat, light concrete finish outside. This structure is light, strong, and of clean finish. See also fig. 137, Askern Airlock.

The Advantages of Airlock upcast Casings are:—(i) Minimum leakage of outside air to the fan drift.

(ii) Casing is not injured badly in the event of an overwind, as in other systems.

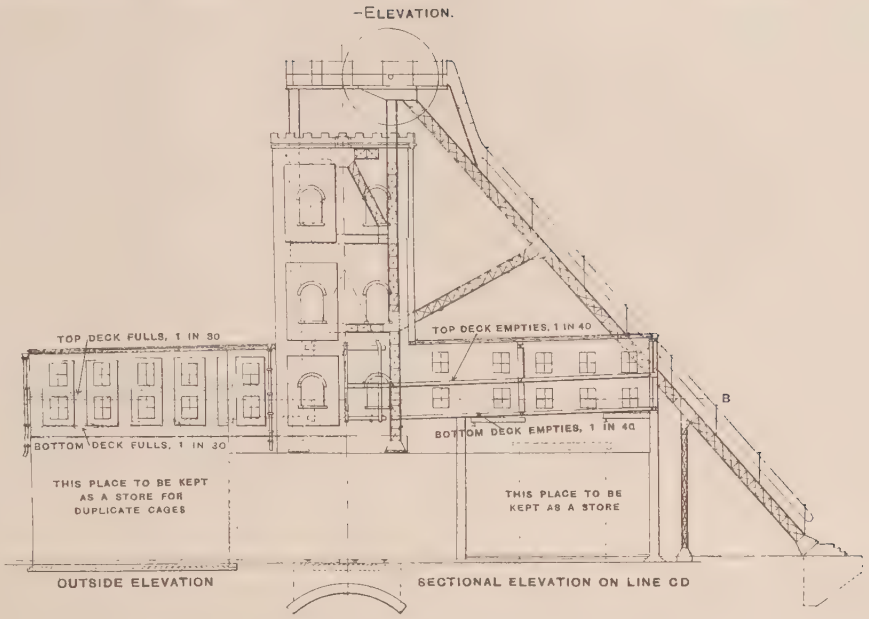
(iii) The dust problem common to some pit-tops is reduced to a minimum.

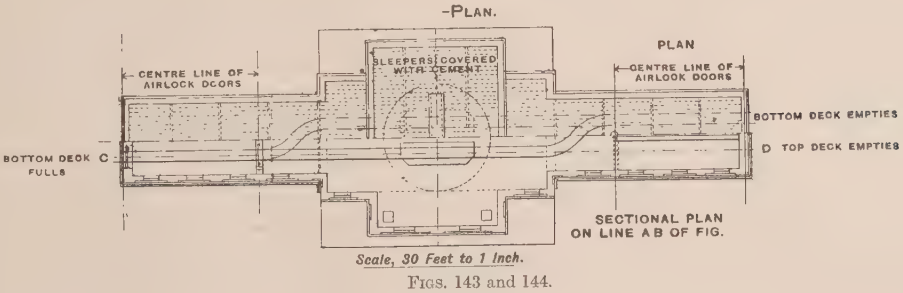
(iv) Supervision and repairs are more easily executed.

(v) The winding rope and bridles are not shocked at each banking as with bonnets or doors.

(vi) The banksman has an unobstructed view of the keps and all the trams about the deck when winding and changing.

Upcast Shaft Top Covers.—The commonest type of banking level covering employed in Great Britain is the boxed-in structure of the type shown in fig. 146. The cage on reaching the bank engages in a pair of bracket arms projecting inward from the bottoms of the two vertical sliding doors. These doors are sometimes fitted with side rollers working on a vertical rail inside the casing or running in a groove. In some cases they are balanced and in others have a retarding device for shock prevention on descent. Such a device we have seen at a certain


—ELEVATION.
TOP DECK FULLS, 1 IN 30
BOTTOM DECK FULLS, 1 IN 30
TOP DECK EMPTIES, 1 IN 40
BOTTOM DECK EMPTIES, 1 IN 40
B
THIS PLACE TO BE KEPT AS A STORE FOR DUPLICATE CAGES
THIS PLACE TO BE KEPT AS A STORE
OUTSIDE ELEVATION
SECTIONAL ELEVATION ON LINE CD
Scale, 30 Feet to 1 Inch.



FIGS. 143 and 144.

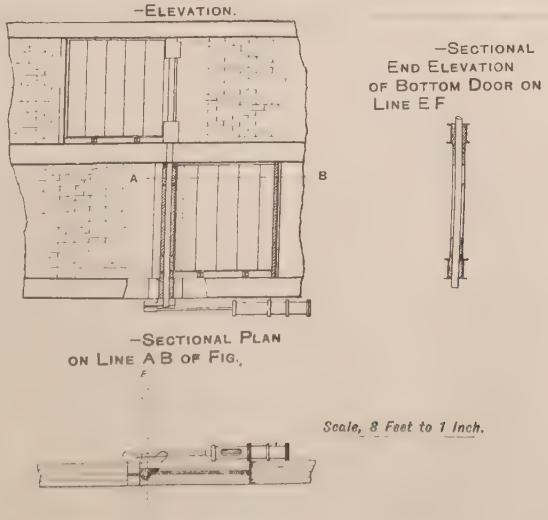


FIG. 145.

Airlock at Maltby Main Colliery

Scottish pit where the doors were attached to a chain passing around pulleys in such a manner that, on the cage descending, the door chains have to pass through a distance equal to the hypotenuse of a right-angled triangle, while the cage passes through a distance equal only to the one side, with the result that the doors descend more gently than the cage and are thereby preserved somewhat.

The casings are made in $\frac{1}{8}$-in. sheet steel or light match-wood,

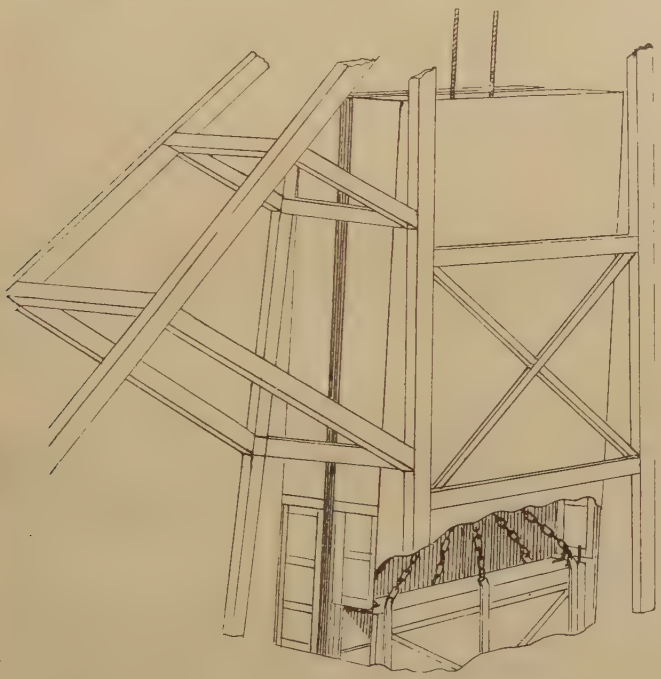


Fig. 146.

and so have the advantage of easy construction and cheapness. They have the following disadvantages :—

(i) Leakage is greater than with an airlock.

(ii) They obstruct the banksman's view of the off side on changing.

(iii) In case of an overwind the casing is seriously torn by the doors.

(iv) On long waiting periods both cages have to be suspended

to make the seal complete; this eliminates much of the advantage of keps.

(v) The winding rope is stressed on the arrival of each cage at bank.

Bonnets.—Fig. 147 shows a bonneted upcast. This arrangement is now passing from the industry because of (i) the stress brought on the rope capel at each banking, (2) the complete wreckage of the bonnet on an overwind, (3) the large amount of leakage. It has the advantages of easy accessibility and supervision, and of cheapness.

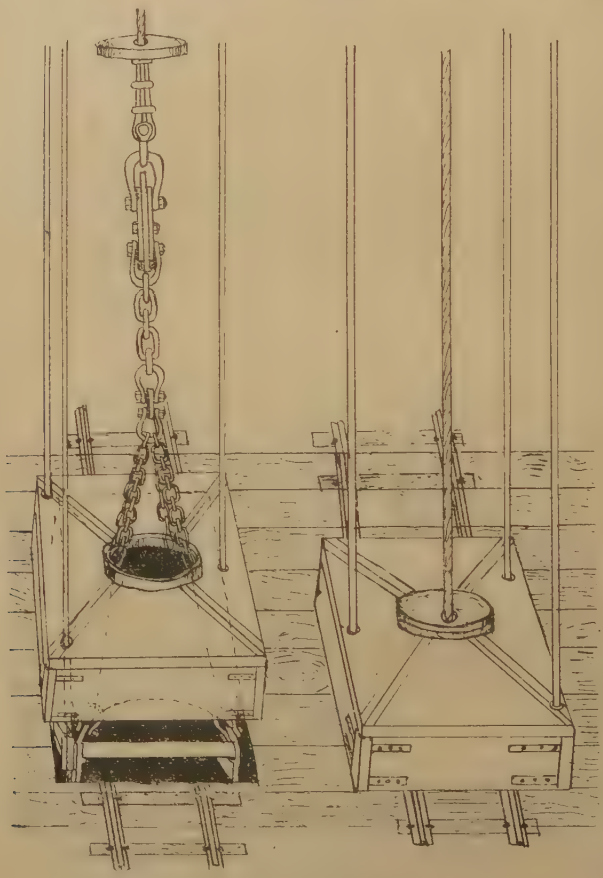


FIG. 147.—Bonnets.

The best form of upcast covering, especially where the upcast is used for coal winding, is the cased-in airlock (figs. 137, 143, 144) previously described. With such a provision there appears to

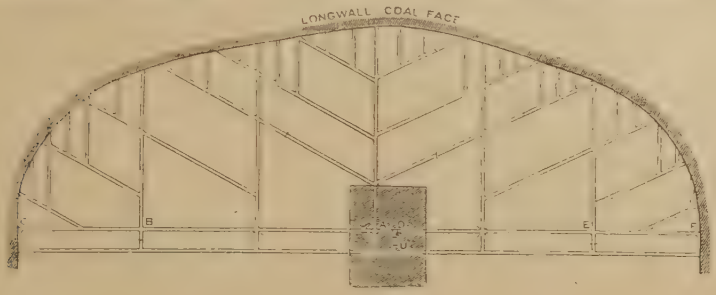


Fig. 148.

be no general reason why the upcast should not be used for winding purposes.

Air Bridges.—These are usually known as air crossings and are employed to carry one current over another, usually return over intake. The C.M.A., 1911, sec. 42 (3), states that they must be so built as not to be liable to be destroyed in case of an explosion. This requirement of the Act has been very vigorously contested by many mining men in the past, who contended that, in the case of a gas explosion, a short circuit of the current due to a damaged air crossing would be beneficial to the workers inbye in that the injurious gases might also be bye-passed. We cannot discuss here the possible advantages to life of a ruined air system in an explosion, nor the similar argument with respect to the inviolable position of the ventilator. We may state, however, that the phenomena of mine explosions (which generally are not gas explosions) go to show that a ruined ventilating system, or wrecked fan, only enhances the dangers and difficulties.

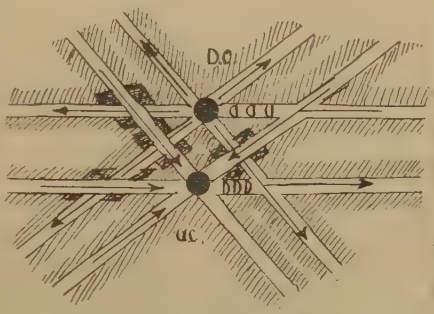


Fig. 149.—Overcasting.

Fig. 148 shows the position of air crossings in a complete side

of a mine worked by the longwall system. The lettered points are stone-dusting machine stations. Fig. 149 shows the intensive application of shaft pillar air crossings.

Air crossings may be either (*a*) natural, (*b*) overcast, or (*c*) undercast.

(*a*) Natural air crossings are those driven in the solid: they have

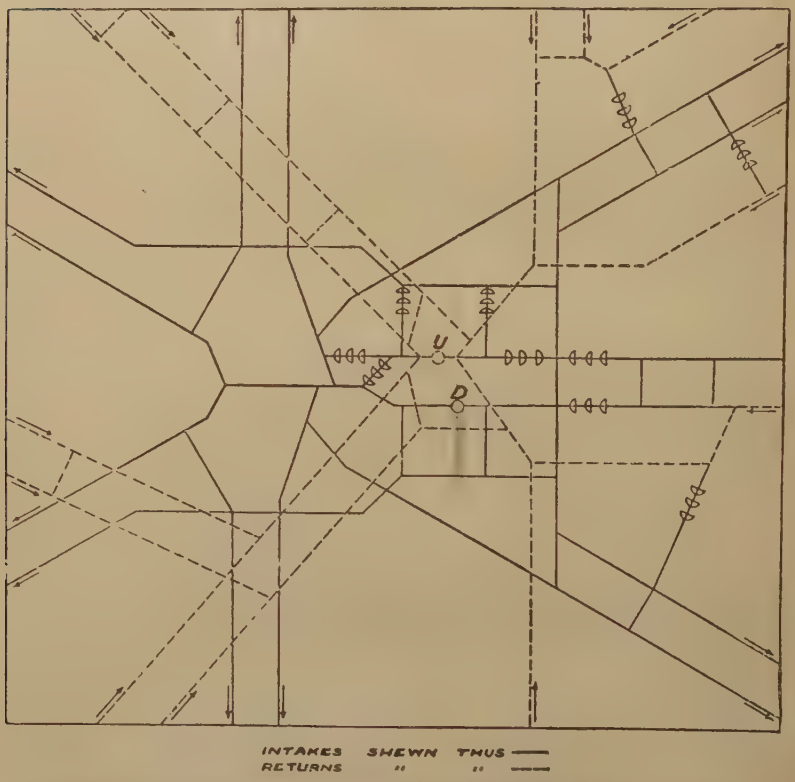


Fig. 150.—Diagram showing method of ventilation at Bentley Colliery.

the great advantage of assured air-tightness which is the first requirement of an air crossing. They are also stronger and, being in the rock, cooler than other kinds of air crossings, and they usually need less repairs. They are becoming very popular, and, indeed, it is thought that in order to combat the temperature problems of future deep mining, that the mains—at least the return mains—will have to be constructed in the solid. An

effective scheme exists at Bentley colliery, fig. 150, where there are no air crossings, each district being supplied by two intakes and having two returns which unite in the shaft pillar and rise to meet the upcast shaft like dumb drifts. This trustworthy and ingenious method of returns, radiating from a point well above the bottom of the upcast shaft, was also decided upon for Askern Main Colliery. Its only disadvantage is that of first cost.

(*b*) Fig. 151 shows an overcast air bridge, *i.e.* one in which the return air is cast over the intake. Of the overcast types, it is the safest, simplest, and usually the strongest type to construct. Many constructional methods are in vogue, including arching, concreting, girdering, walling as in fig. 151, and timbering, all of which are governed by local conditions. The return air being the lighter, should always pass over the intake as shown in fig. 151. Here and in fig. 152, A and A′ are light **I**- or **H**-section rolled steel girders about 6 in. by 5 in., set their own width apart on an oak seating and sometimes concreted between, as in fig. 152, though this latter practice hinders renewal and repairs. It is customary to cover the girders with 3 in. by 9 in. planking, B, and these by concrete, grouting, and earth, as at E. Nine-inch to 18-in. side walls, as at D, are commonly used, the whole type being very popular owing to its simplicity, strength, and ease of construction.

(*c*) **Undercasts** have the disadvantage, as will be seen by fig. 152, of having a secondary floor for the intake, which is liable to distortion with weighting or heavy traffic and so cause leakage. Furthermore, the lighter air has to take an unnatural path under the intake. Lighter air has, of course, a tendency to rise. Again, the construction is usually more expensive. Small undercasts are often easily and cheaply constructed by sinking a Lancashire boiler flue tube into a prepared trench cut across the intake roadway floor. The dropping of portions of large boiler shells for undercasts makes a splendid job, but needs great skill and much space in order to overcome the transport and fixing difficulties.

Ventilation Plans.—The legal requirements with respect to mine ventilating plans are stated in the C.M.A., 1911, sec. 20 (2), and it is there specified that a separate ventilation plan must be kept at the mine showing the system of ventilation, direction of air currents, the points where the air is measured, and the principal accessories to ventilation. Also the intakes must be coloured blue and the returns red. Such a plan must be prepared by a

qualified surveyor and be made on a scale of not less than 40 in. to the mile (2 chains = 1 in.), and in the case of a mine opened before 1911, the scale may be not less than the $\frac{1}{2500}$ Ordnance

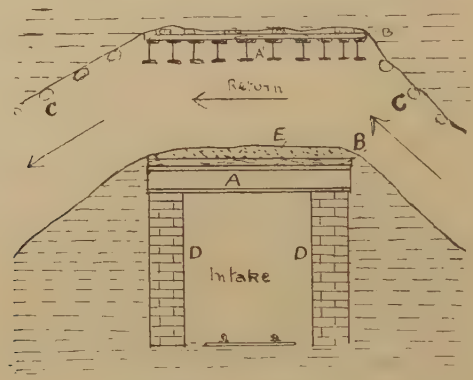


Fig. 151.—Overcast.

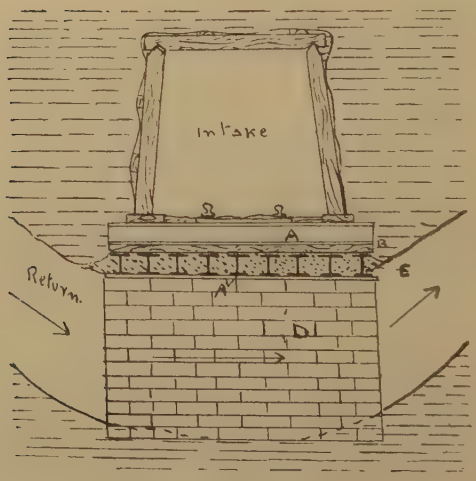


Fig. 152.—Undercast.

scale, or 25·344 in. = 1 mile. The owner, agent, or manager shall always give an inspector access to such plan, and on request mark such points thereon as the Act stipulates. The inspector may, for official purposes, request a copy of the plan to be made,

and the said mine officials will be held guilty of an offence against the Act if they fail or refuse to do so, or if they conceal any portion or produce an incorrect plan. In case of such offence the inspector may require, by notice in writing, a new plan to be made at the expense of the management within twenty days or

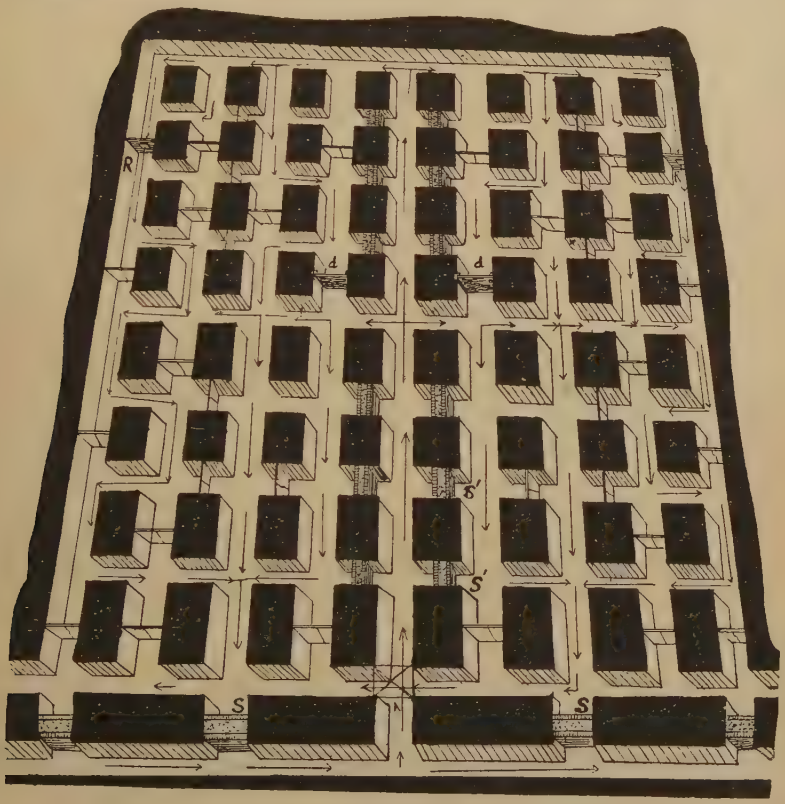


FIG. 153.—Ventilation of a gassy panel.

such time as the Secretary of State allows. If this is not complied with the management is again liable. Another enactment which greatly affects lay-out, and consequently benefits ventilating, is that of sec. 42, wherein it is required that for every seam in a *newly opened* mine two main intake airways must be provided, but not necessarily driven in the seam itself. One of these roads must not be used for haulage. There are several cases whereby exemption can be claimed, but what chiefly concerns us here

is that the adoption of this enactment means in many cases the carrying forward of three parallel main roads, since haulage may claim two and one must not be used for haulage. By a main airway is meant, legally, an airway commencing from, or terminating at, a shaft or outlet to the surface, or any airway from which a split is taken to ventilate any district of a mine (see sec. 122, C.M.A., 1911). Any part of a seam having an independent intake airway commencing from a main intake airway and an independent return airway terminating at a main return airway is legally considered a "ventilating district."

Fig. 153 shows, pictorially, the position of the ventilating accessories in a completed panel ready for retreating in a very gassy seam. The regulators at R R may be withdrawn outbye a portion of the distance, or entirely if there is an excessive short circuiting at the main split. In such case the doors *d d* will have to be re-arranged on the outbye side of the central lateral splits. The main stoppings at S S, etc., and the temporary ones at S′ S′, etc., should be all brick-faced for about 9 in. deep and filled with ashes or puddled clay between the walls, clay being preferable for permanent stoppings.

Fig. 154 shows a proposed method for the ventilation of a hot and gassy mine such as might be expected in water-sealed strata in mines under the sea, deep mines, etc., by the longwall method in accordance with sec. 42, C.M.A., 1911. In such cases the strata should be fully packed, the shaft pillar perimeter packed, and the maximum amount of air taken direct to the face. The figure shows one side of a symmetrically developed mine, the downcast being shown by a white and the upcast by a black annulus. The fresh air, from the former, splits at the shaft pillar edge to form the two main intakes A and A′, the latter not being used for haulage. A″ is a main haulage return provided with an emergency door at C, which may be opened if the leakage of air past the sheets B′ and B″ enables sufficient air to pass to adequately cool the main haulage pillars and also keep up the return face supply. The air crossings on the mains from the downcast, at E and E′, may be of the ordinary natural type in the solid with provisional separation doors beneath, in which case there will be auxiliary separation doors at S on either side of the pit, and later, as the mine develops, at S′. Or the overcasts may be avoided by drifting from the downcast, a few hundred feet up above the bottom, radiating intakes, and similarly radiating the main returns up to the upcast shaft some distance above

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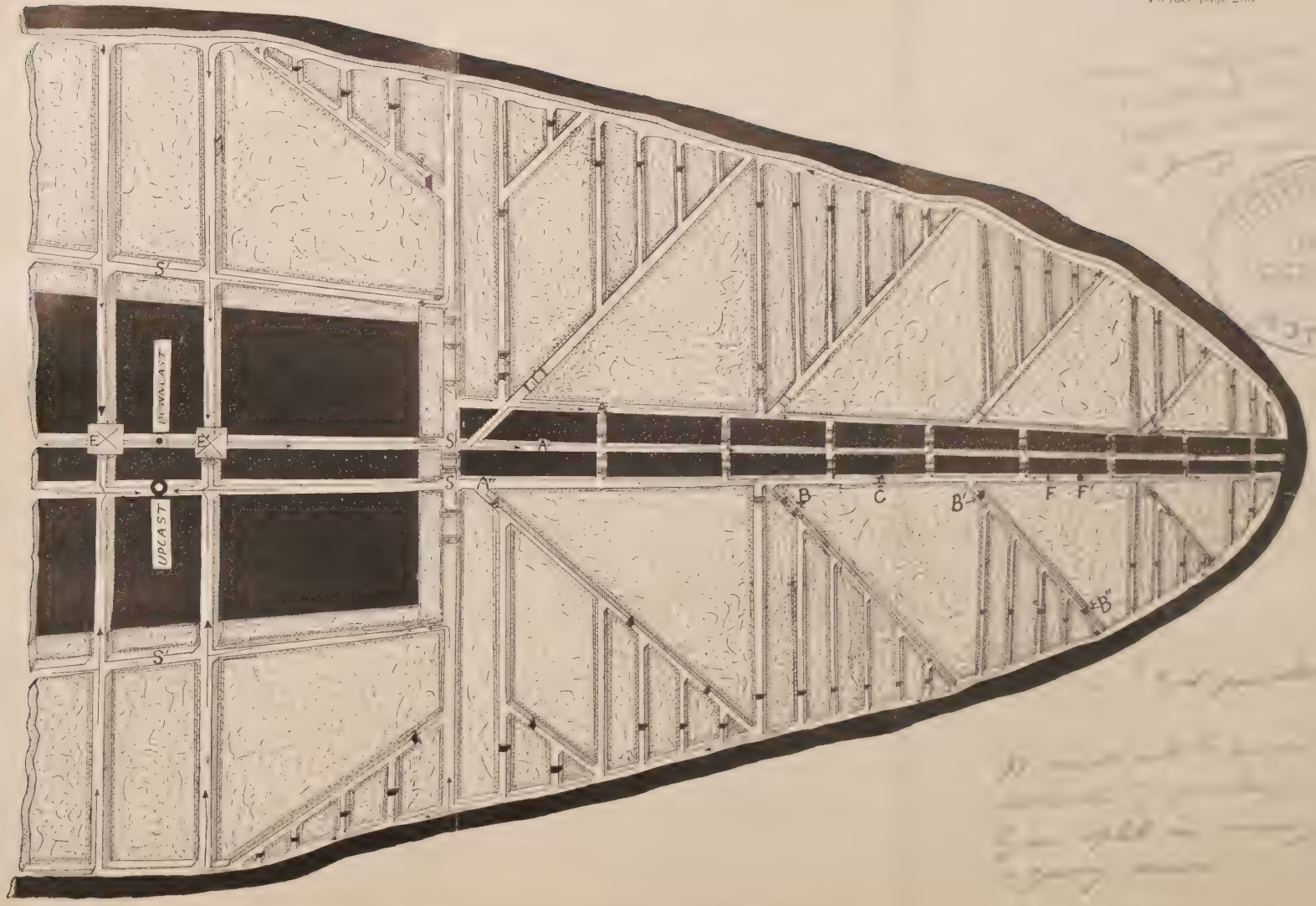

FIG. 154.—Ventilation of a hot and gassy seam.

to the last bottom. The main coal support should be in the form of pillars, for coolness, each pillar having a stopping to its fellow, but the air coursed through the last few pillars to avoid excessive dust on the faces. The doors F F' move up as each pillar is stopped to its neighbour. The return haulage engine should be in the intake: the practice of split-heading an engine-room in the return is very objectionable.

Ventilation Plans. Among the following plans will be found several from the various First-Class Colliery Managers' Examinations set by the Board for Mining Examinations since 1911. Many of these plans leave some things to be desired; for instance, the direction of dip and cleavage are not always mentioned, and both are very useful factors to be minded of when designing the ventilation of a colliery on a plan. Again, the candidate's initiative and originality are damped somewhat by the requirements of the examiners, who in most cases definitely state how many splits, air crossings, etc., are required. This is unfortunate, since the plans are such splendid practical examples, and in some cases taken from the subjects of legal inquiries.

In a large measure the conditions deciding the layout also control the method of ventilating the workings, so that we may state, practically, that there is a pillar and stall and a longwall type of ventilation. A careful study of the prevailing circumstances must first be made, since each mine differs from any other mine. In order to guide the reader into the most desirable paths in the ventilation of set plans or prospective layouts, we will venture the following *fourteen points of plan ventilation*.

(i) Split the air judiciously, making the main splits as near the downcast bottom as possible and the returns as near the upcast bottom as possible. Make all the splits equal, or as nearly so as practicable, in order to equally apportion the air.

(ii) Avoid excessive splitting and also long coursing.

(iii) In a fiery mine, or one with a friable coal roof, or where there are gassy pillars, convey a stronger current to the face than in normal cases. This often means duplicating the main fan, installing an auxiliary one, or speeding up.

(iv) Avoid installing many air bridges or many regulators. The former often means increasing the tortuous nature of the airways, the latter wastes quantity. Avoid air crossings in the "weighting" zone. We would suggest a quarter of a mile as the minimum distance from the face in a developed mine.

(v) If possible convey the fresh air direct to the lowest part

the upcast bottom. The main road support should be in the form of pillars, for coolness, each pillar having a stopping to its fellow, and the air coursed through the last few pillars to avoid excessive dust on the faces. The doors F F′ move up as each pillar is stopped to its neighbour. The return haulage engine should be in the intake : the practice of split-feeding an engine-room in the return is very objectionable.

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(i) Split the air judiciously, making the main splits as near the downcast bottom as possible and the reunions as near the upcast bottom as possible. Make all the splits equal, or as nearly so as practicable, in order to equally apportion the air.

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(v) If possible convey the fresh air direct to the lowest part

of the workings, so that when rarefied, by increase of temperature, it may ascend to the rise workings, assisting the ventilator in so doing.

(vi) If a regulator must be used, place it behind the last working place on the return side in the district affected.

(vii) Avoid doors on main haulage planes.

(viii) If an auxiliary fan is being used, place it as near the point of zero water gauge as practicable.

(ix) Air lock the separation doors and upcast casing if possible.

(x) Ventilate, if practically possible, on the line of cleavage; the friction is far less. Hang sheets, doors, etc., on the windward side of the bars or posts.

(xi) Always try to apportion the districts on a mine plan so that small districts may be quickly and effectively isolated in case of fire, etc.

(xii) Avoid strong opposing currents in close proximity; this often means the installation of stoppings in places where such currents are inevitable. The chief danger here is leakage.

(xiii) A long straight face or district with uniform panelling is easier to ventilate than broken-face lines and ribs.

(xiv) If it does not pay to keep good ventilating arteries, then bad ones are out of the question.

The ventilation of set plans or layouts being governed so much by local conditions, output, and other physical factors, is a very debatable subject. In our remarks applying to the accompanying plans the following assumptions are understood, except where the requirements or layout are to the contrary:—

(*a*) That the workings are flat.

(*b*) That the upcast is a coal-winding shaft.

(*c*) That straight up the plan (normal to the lettering) is the direction of north.

Note re *plan, fig.* 155.—An alternative scheme to the one submitted would be to have an extra air-crossing at A (fig. 156), a regulator at B and one at G (fig. 155), and the current reversed in the B road, air coming from the north level, the doors at C being abandoned. This idea has not been adopted owing to the expense of the extra air crossing and six doors being required —three at D and three at E—and the wastage due to regulators. There is a regulator at R in the S.E. district.*

Note re *plan, fig.* 157.—This plan, with the S.E. district very

* This plan is given as that of the West Riding Colliery, Yorkshire, which exploded in 1886.

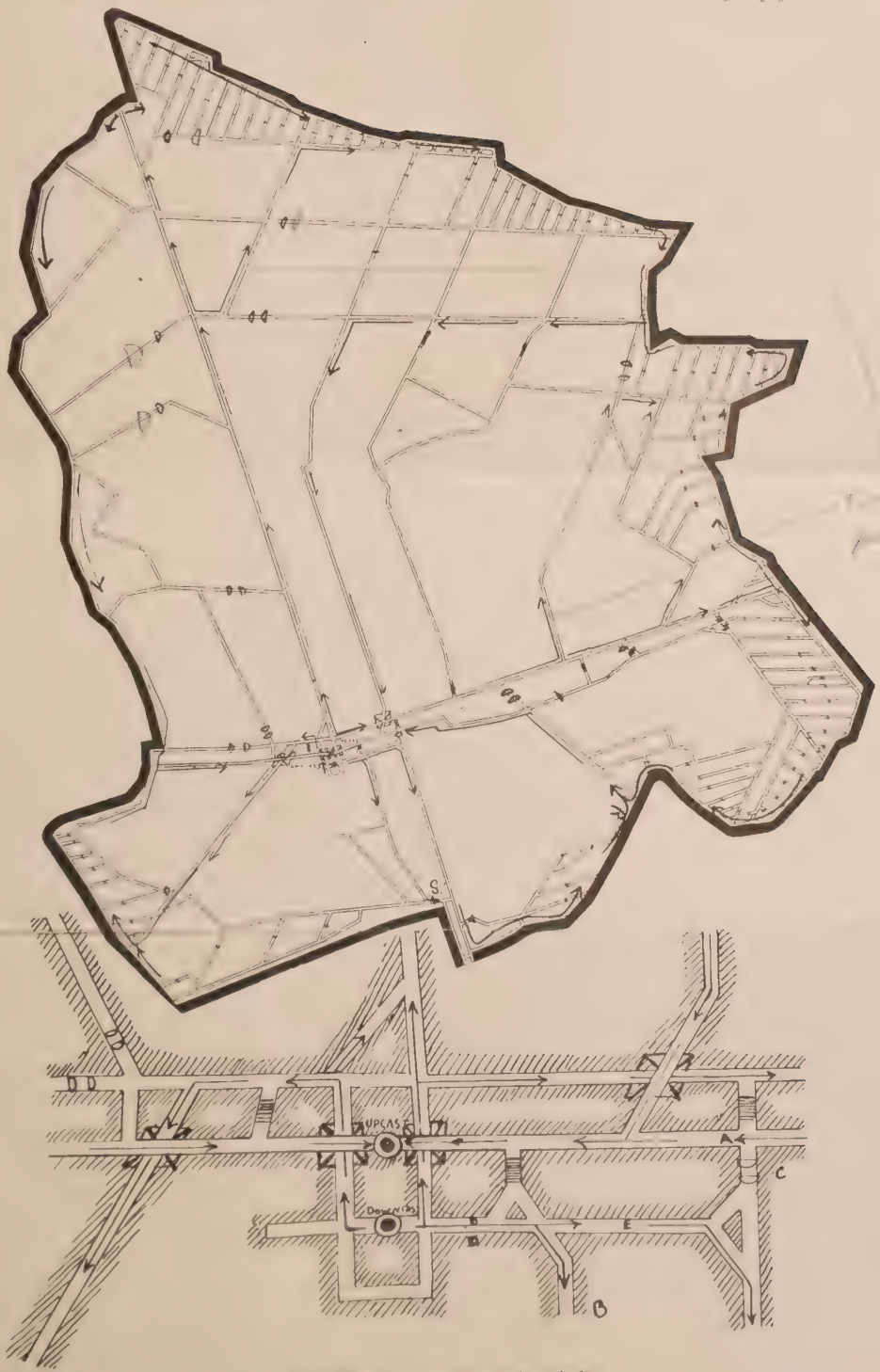

FIG. 155.—Downcast the winding shaft.
FIG. 156.—Enlarged plan of pit bottom.

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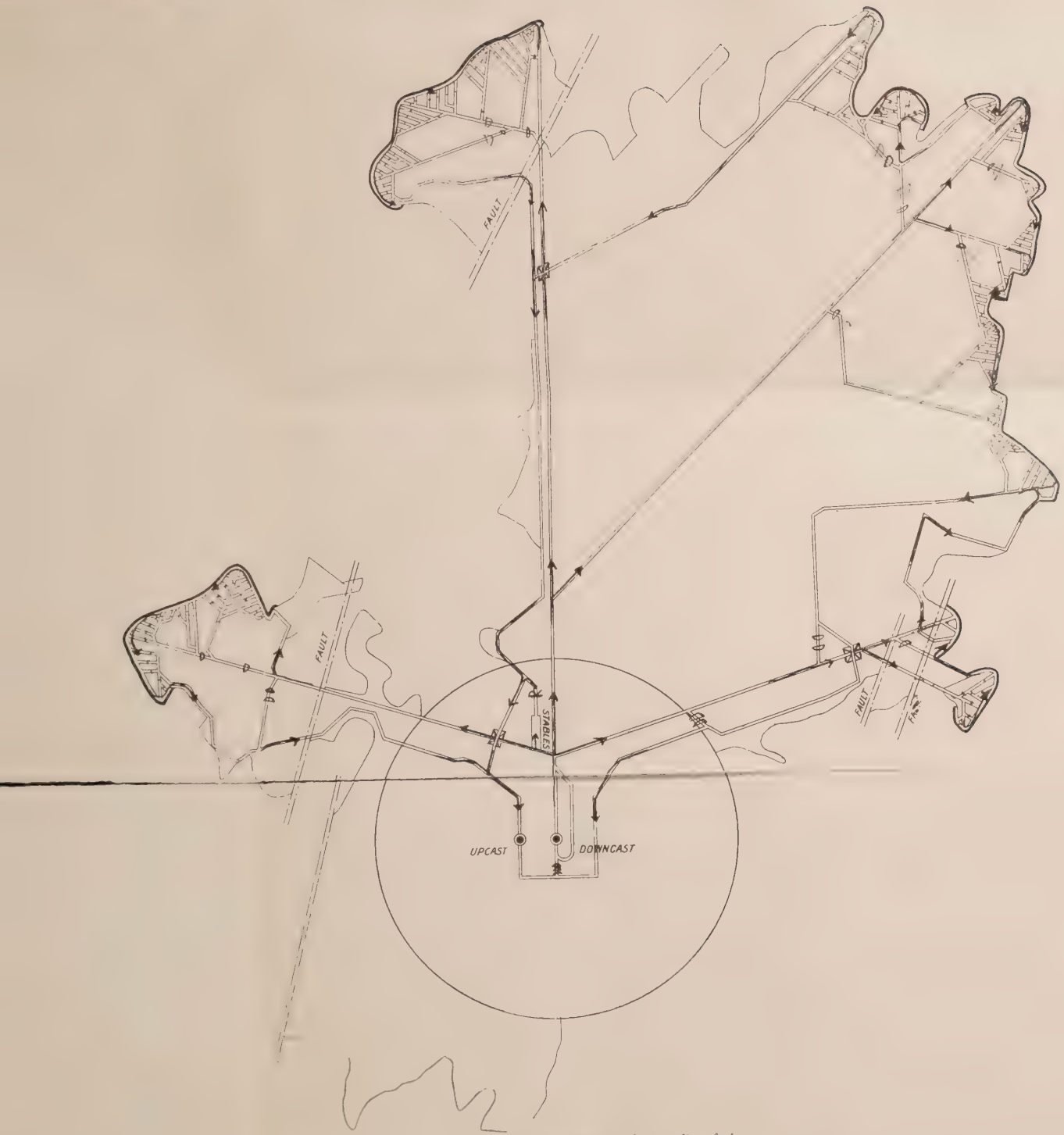

Fig. 157.—Ventilation with six splits of air.

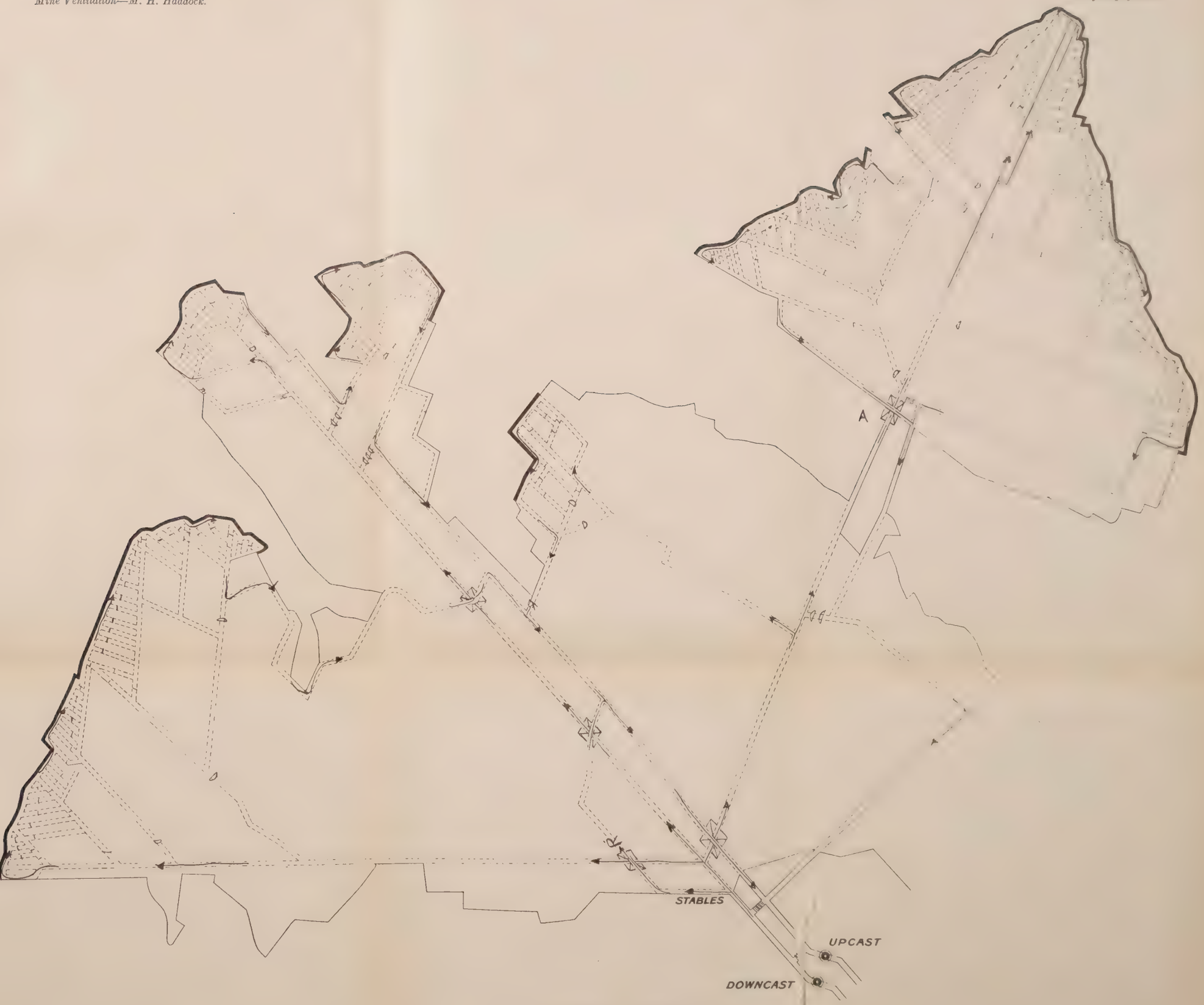

FIG. 158.—Ventilation with six splits of air.

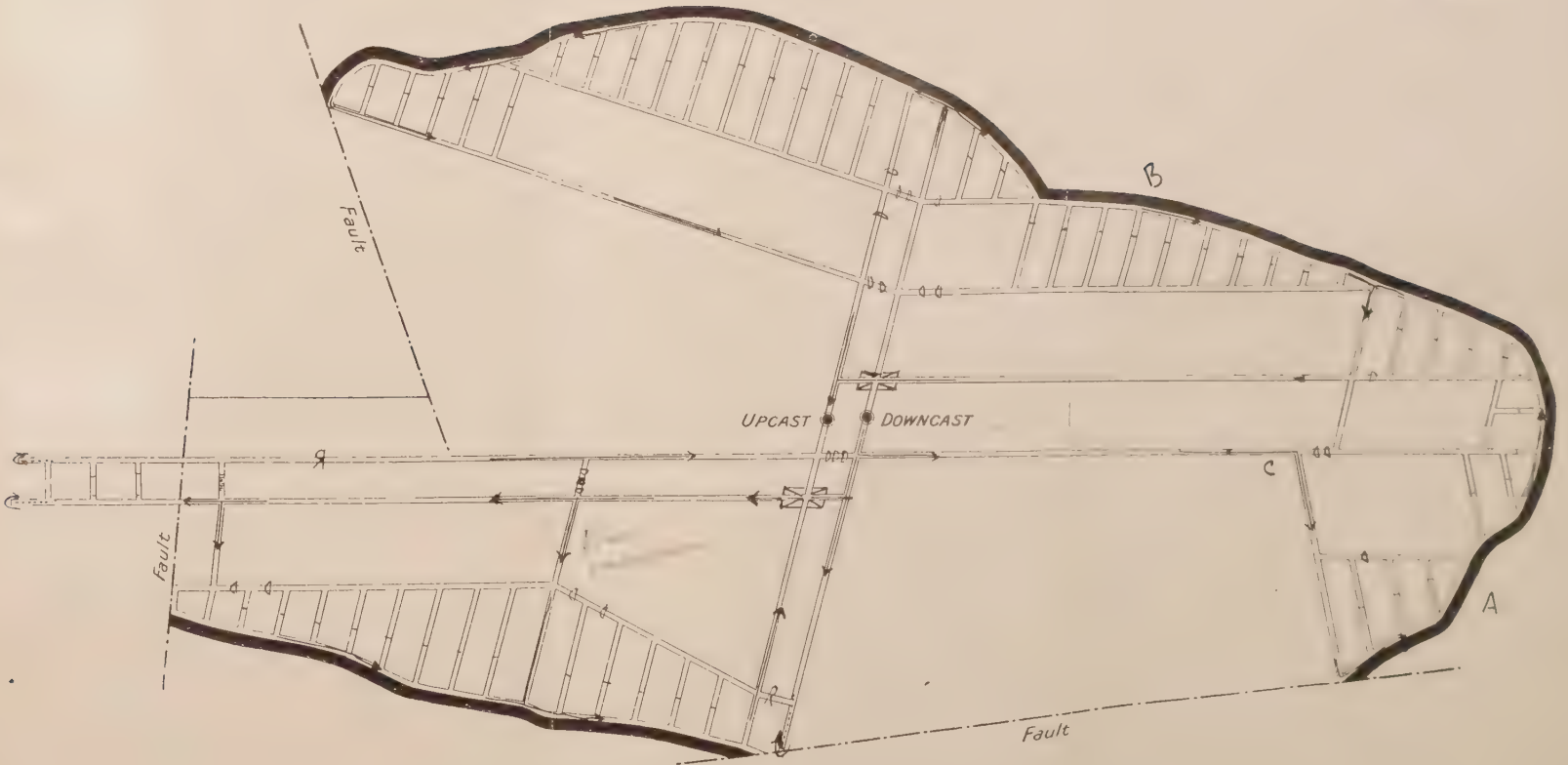

FIG. 159.—Ventilation with five splits of air and two air crossings (downcast the only winding shaft).

FIG. 160.—Ventilation with five splits of air and four air crossings.

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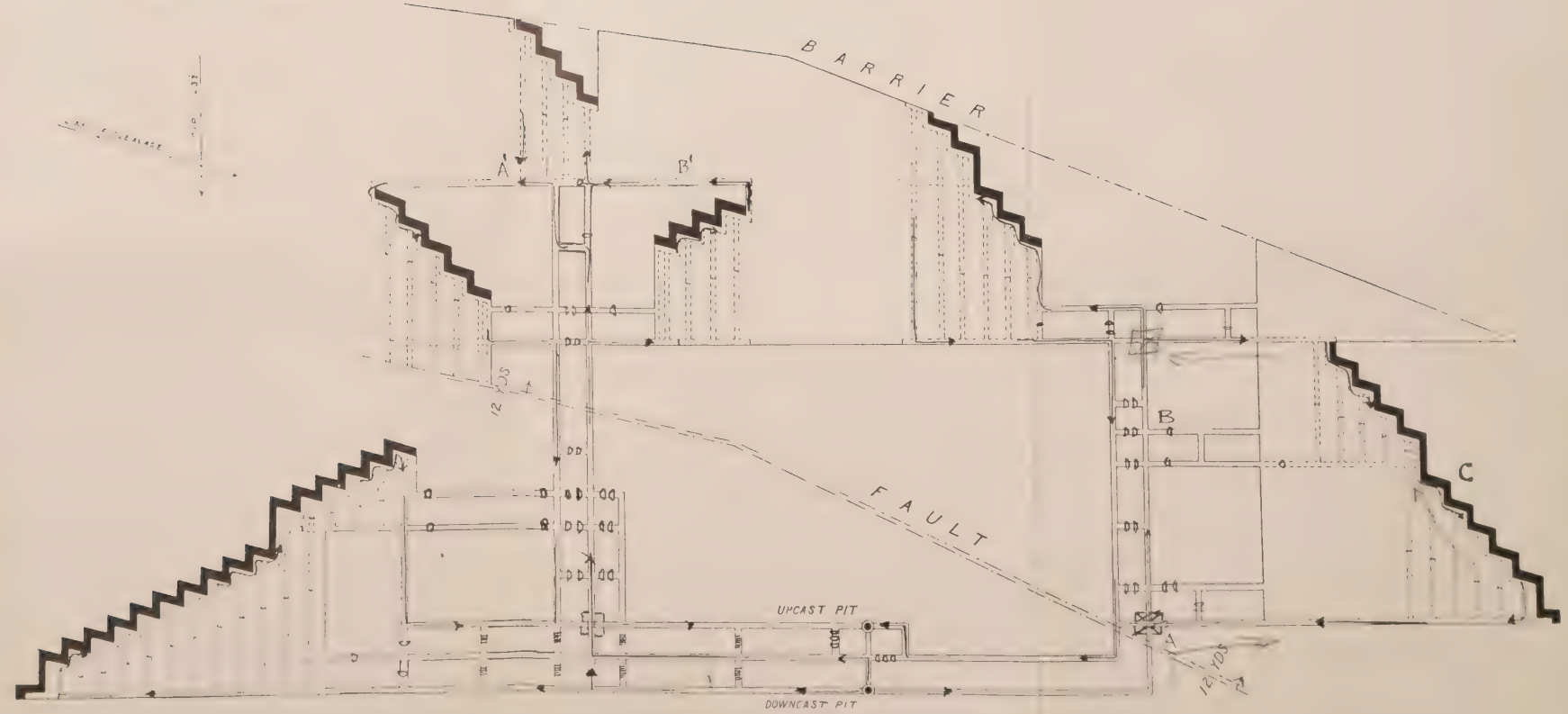

FIG. 161.—Ventilation with four splits of air and two air crossings.

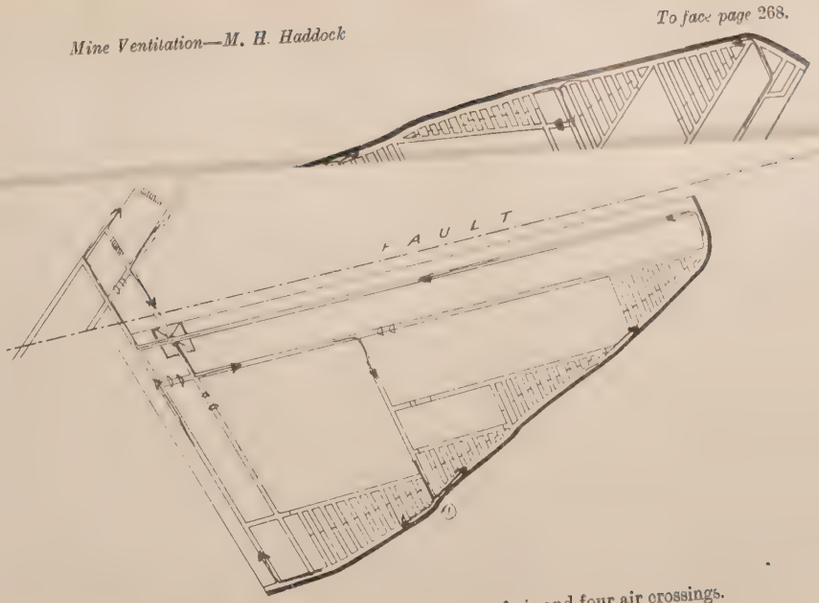

FIG. 162.—Ventilation with five splits of air and four air crossings.

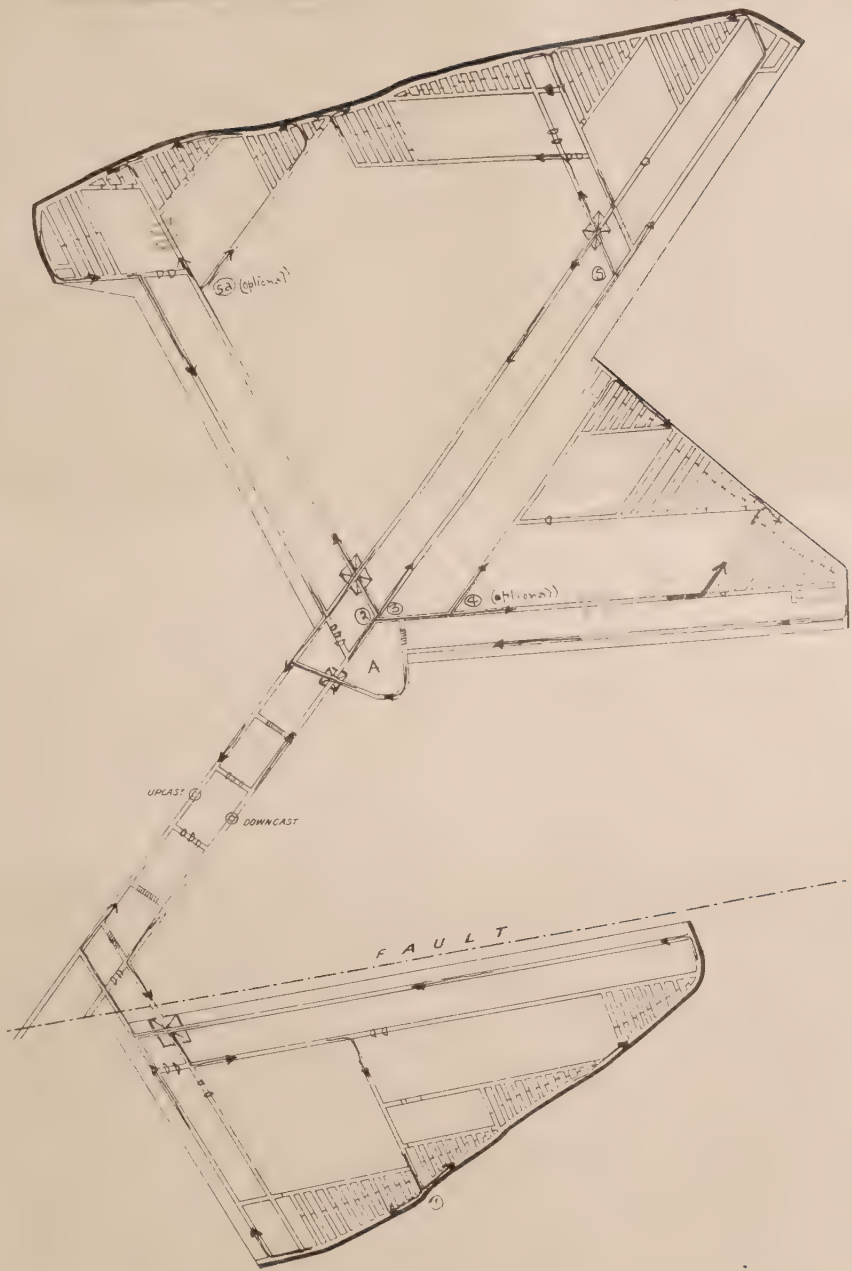

FIG. 162.—Ventilation with five splits of air and four air crossings.

slightly more developed, and without stables, was again set by the First-Class Colliery Managers' Examiners in November 1917. Stables conform with the legal requirements of the C.M.A., 1911, respecting a district. All pit-bottom stables should have an independent regulated split.

Note re *plan, fig.* 158.—The air-crossing at A is in a position where there is a liability to weakness owing to the number of roads meeting in the vicinity, and also the traffic. Its abolition or removal would necessitate very awkward and tortuous coursing, and would also affect the west-side scheme. An alternate scheme will save an air crossing, but provide more trouble: shaft-pillar air crossings are usually permanent, needing very little attention. This plan reappeared at the May 1918 Examination, and a modified form of the N.E. district in May and November 1922 Examinations.

Note on plan, fig. 159.—The current on the east main intake is diverted at C in order to apportion equally the quantities and pressures in districts A and B. The developing heads on the west levels and the head proving the fault on the south main are brattice-sheeted to the face and regulated. This plan belongs to a similar series which appeared in previous and subsequent examinations, slightly modified.

Note on plan, fig. 160.—The air crossings are at their maximum distance inbye in the N.E. districts, because the uniform development of the face about the fault in their proximity precludes their removal to any point nearer the faces. This development will, in time, "weight" the junctions further inbye and so cause leakage at any air crossing placed at such points. This plan, with a S.E. district added, appeared in the Colliery Managers' Examination of May 1916, when seven splits were required, and again in May 1919, when five splits were specified; in the latter case the plan was exactly similar to fig. 160.

Note on plan, fig. 161.—The air crossing at A, though in the best place for symmetrical apportionment of the air on the east side, would be better placed at B, and the air face-coursed around district C up to it, if there is sufficient ventilation to place output first. Coal, with the air-crossing at A, would have to travel under and over it if both shafts are winding coal, which means a heavily stressed air crossing and an expensive construction. The reunion at A′ could be obviated by a regulator at B′, but the districts are too small to warrant special subdivision.

Note on plan, fig. 162.—In the north and east districts the

method of ventilating by augmented splits is shown here as an example. This will make really six splits, a full coursing in the E. or N.W. districts will give five. This method cannot be applied in gassy workings where the products of one face would be taken along another. The system of ventilation is very materially altered by assuming that A is a stable split. Counting the main division at the pit bottom as a split, both optional splits can be discarded, and their districts face-coursed. This method of layout and ventilation is popular in seams liable to spontaneous combustion owing to the ease with which the isolation of single districts can be effected in case of fire.

Figs. 163 and 164 are plans showing the early stages of a modern mine development wherein output and ventilation are equally important. The former shows rapid early development with machine coal-cutters, and the latter a proposed method of opening out in very deep mines or other places where the "weighting" is great. Both figures are hypothetical and are left unmarked in order that the reader may exercise his skill in plan ventilation. Considerations of space prevent us adding several more such figures at this point.

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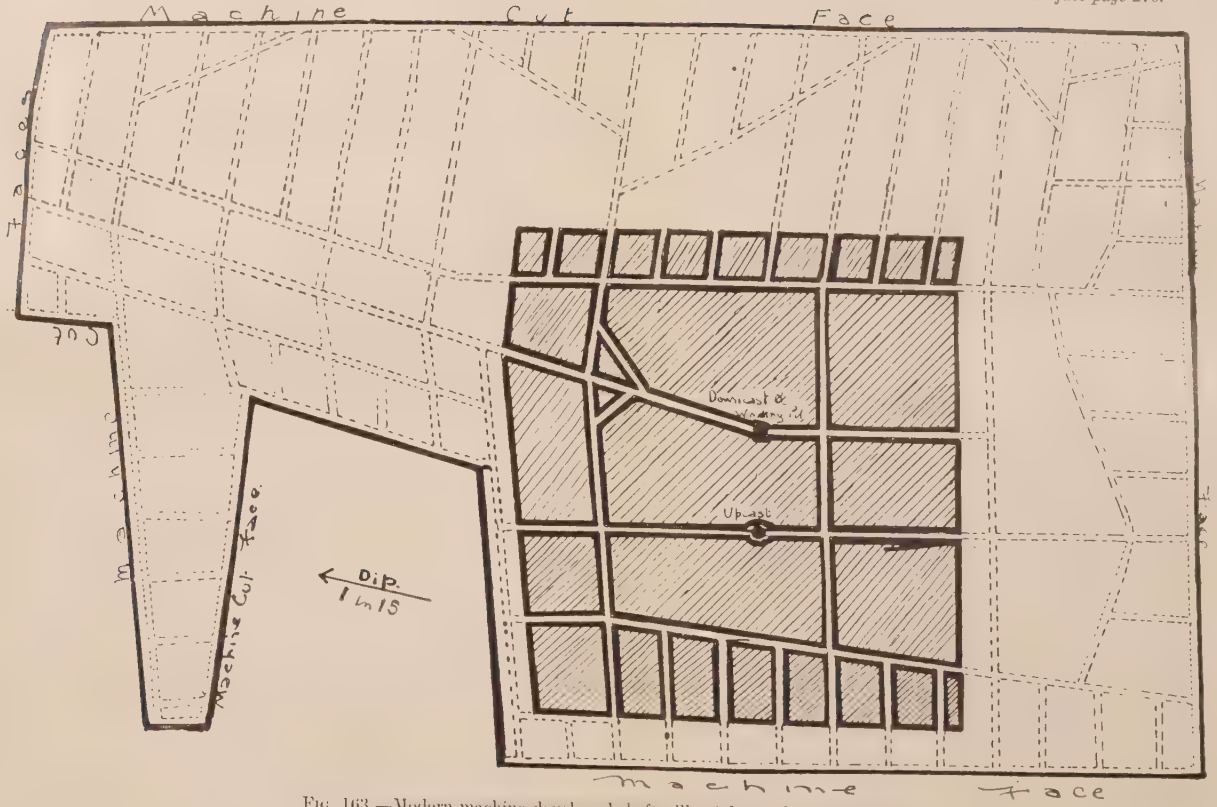


FIG. 163.—Modern machine-developed shaft pillar (plan to be ventilated).

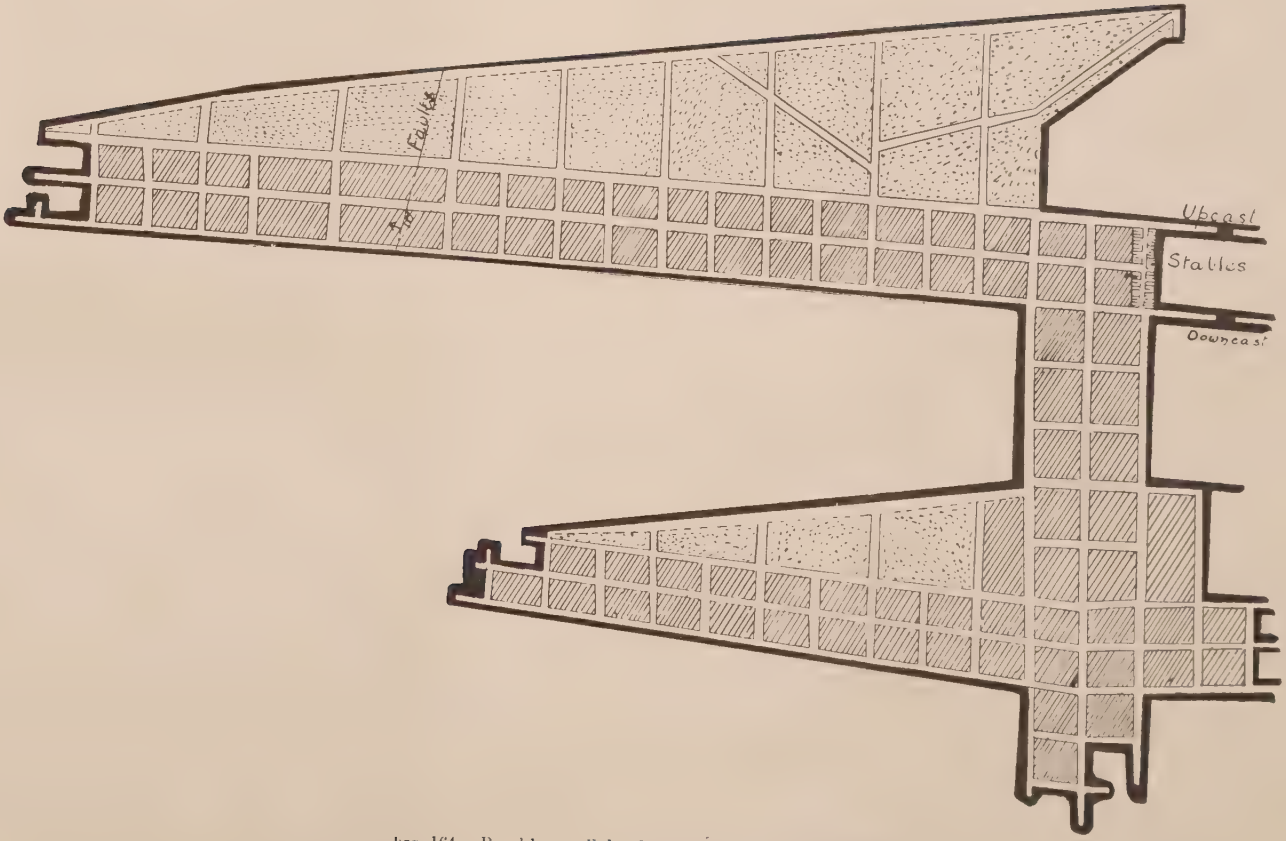


FIG. 164.—Panel-longwall development (plan to be ventilated).

CHAPTER IX

TYPES OF CENTRIFUGAL VENTILATORS

The number of types of ventilating machines in use in various countries is almost legion, therefore we will restrict ourselves to the chief kinds used in British coal mines. They are best classified in respect to the style of blading.

(i) **Backward-bladed Fans.** -Blades of this type have the advantage of reducing peripheral eddies, but they require a greater speed to produce a given head, and are consequently higher in mechanical friction than other types. They seldom show a manometrical efficiency above 56 per cent.

The angle of curvature of the blades in all fans is seldom more than 45 degrees, and in many fans the greater part of the blade length is straight, the curvature being only at the tips. Most modern fans only employ the tips.

The Guibal Fan.—This is the pioneer centrifugal fan, and it takes its name from the celebrated Belgian engineer-professor. It consists of an octagonal drum, to each face of which is bolted a straight arm carrying a match-board sheet or vane across its extremity, so forming an air paddle, and lying back from the radial line. The fan can be arranged for single or double entry of air. The evasée expands only on the off face, the flare being about $7\frac{1}{2}$ degrees, the near face being vertical. It has the same width at centre and circumference, which causes it to deliver the air more rapidly than it can receive it. This is a disadvantage, because it causes eddying in the delivery passages, and to obviate this drawback a sliding shutter, D (fig. 165), working off the pulleys F and F′, is fitted. This shutter regulates the cut-off ordinate to meet existing conditions.

The Guibal fan possesses the following merits :—

(i) Being of large diameter it is slow running, thereby minimising the risk of breakdown.

(ii) When in good order it is capable of dealing with fairly large volumes of air (this advantage is common to most fans for mines).

Its demerits are :—

(i) It is more costly than smaller fans and occupies more space.

(ii) Owing to its great mass (up to 40 ft. in diameter) the foundations needed for it are more expensive than for small fans.

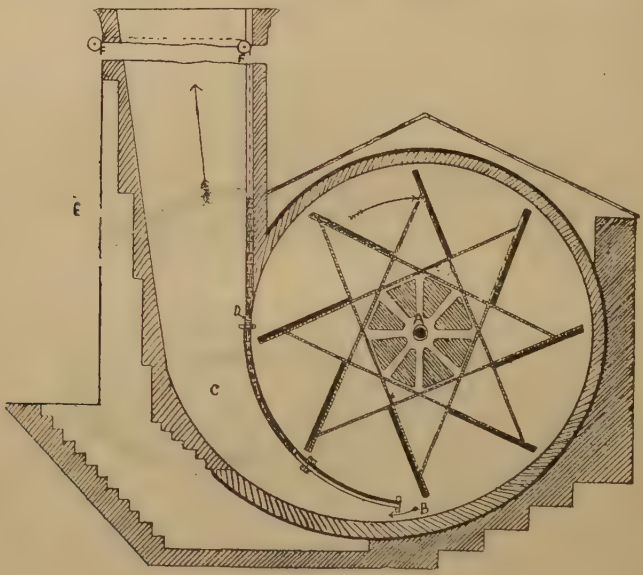


Fig. 165.—Guibal fan.

(iii) The efficiency of heavy fans is less than that of lighter ones owing to the greater driving power needed.

The manometrical efficiency of the Guibal is about 45 per cent., but, apart from this and the above demerits, they hold their own fairly well with other types.

All boss-bladed fans, *i.e.* those which have their blades attached to the hub by a spider, are less in width than in diameter. The width is usually such that the cylindrical surface swept over by the inner edge of the blades is approximately equal to the inlet area. This, it is thought, prevents excessive radial velocity, but the exterior profile of the fan also considerably decides this factor. They seldom have a greater area

Fig. 166.

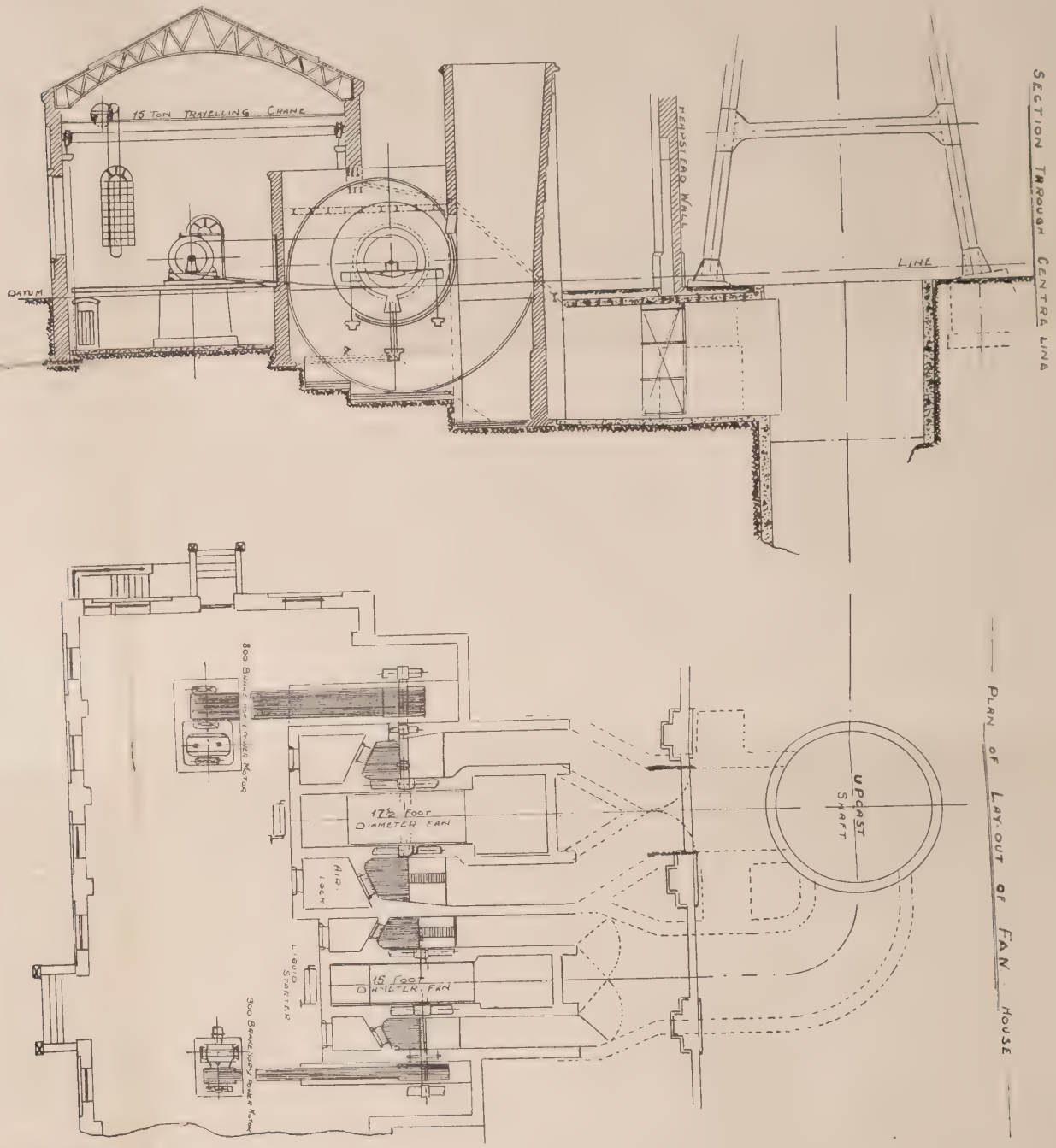


Fig. 167.

Installation of Walker fan at Blackhall Colliery.

of inlet than half that of the fan side. If the blades taper the circumferential width of them is about half their inlet width.

The Walker Indestructible Fan.—The popularity of this fan depends upon its strong and reliable construction, which enables it to run for considerable periods with very little supervision and repairs. The runner has eight to twelve blades, see fig. 83, which are narrowed at the centre and are bent backward from the direction of rotation. They are braced to the central boss by angles and webs, and the whole is of steel throughout. The fan is made in sizes up to 18 ft. diam., and of speeds of from 120 to 300 r.p.m. The V-shaped shutter on the near face in the evasée can be raised or lowered at will in order to vary the load on the vanes and so lessen vibration. The manometrical efficiency is about 60 per cent., and, after an all-round consideration of all its features and performances, it can be safely classed as one of the best of the heavy-bladed types.

Figs. 166 and 167 show the Walker fan installation at Blackhall Colliery, Durham,* the smaller fan being used as a standby.

A disadvantage common to all fans with few blades is that "the proportion of the air particles in actual contact with the vanes at the instant of discharge is small compared with the total number of particles discharged; and in proportion to their distance from the vanes at the instant of escape from the fan-runner the particles will attain to an increasingly less tangential velocity than that indicated by calculation." † This decrease of tangential velocity will be greater the smaller the fan-radius, so that for a given tip-speed the virtual slip of the air-particles will be twice as great for a runner of one half the radius. The effect of this virtual slip is to decrease the manometrical efficiency, *i.e.* to prevent the fan from giving the full theoretic depression. A greater depression will be given by a larger fan than a small fan running at the same peripheral velocity, hence the increase in diameter for increasing duties.

The Waddle Fan.—The Waddle fan is the only open-running

* By the courtesy of Messrs Prest & Leggatt, Horden Collieries, Durham.
† Mr Joseph Parker, *Trans. Inst. M.E.*, lxiii. 226.

fan used in British mines to any appreciable extent, the Moritz, an American open-running fan, being only used to a very limited degree in Europe. The older Waddles were made in sizes up to 45 ft. diameter and running at speeds rarely over 150 r.p.m. The modern Waddle has a diameter usually within 18 ft., and runs at speeds up to and over 300 r.p.m., and has its vanes specially turned near the inlet. The fan takes air only on one side and consists of two parallel sheet-steel side-discs, seldom more than 40 in. apart, and braced together by the vanes, which are riveted to them by webs and small gussets, the whole revolving together. The general lie of the blades is backward in a curve of about 30 degrees from the radius line, and their number from eight to twelve. A hollow conical centrepiece allows the end of the bearing to project well through the inlet inside the centre of gravity of the fan, so that the weight comes directly over the bearing. Such an overhung bearing is shown in fig. 120. The blades of the modern Waddle have a wedge-shape narrowing outward from the single inlet to their peripheral edge, which is short of the actual disc edge by 12 to 20 in. according to size. The effect of this narrowing is to keep the velocity of the air throughout the fan in strict ratio with the centrifugal force, so curtailing re-entry of air at the tips. The effect of the projection of the side-discs beyond the blades, and their outward curvature at this point, is to transform the kinetic energy of the outgoing air into potential energy, *i.e.* an evasée effect.

The features which make the modern Waddle fan an excellent and popular ventilator are its strength, compactness, short shaft, simple masonry, single bearing, and adaptability to direct drive.

> The effect of gradual forward curvature of the blades at the inlet edge is to give an easy sweep or passage of the air on to the blades. This scoop has been subject to much controversial discussion, many holding that it increases the internal friction.

The Chandler Fan.—This fan, made by Messrs Bumstead and Chandler, of Hendesford, belongs to the modern type of short-bladed machine. Its blades, numbering twenty to thirty, are curved backward at the tips to give an easy delivery. They project beyond the bracing annulus which bounds the inlet edge. The fan may be made with single or double inlet, and, like all short-bladed ventilators, has a large inlet, which here is about 70 per cent. of the end area. Very few shallow-bladed fans

have backward-inclined blades: the makers claim that a more consistent and positive depression is created by their arrangement. This will be seen to be a practical advantage in very dusty atmospheres: a shallow, forward-bladed fan in such a case often collects dust in the hollow of the blade curve, so that in time the advantage of such curvature disappears, the blade being

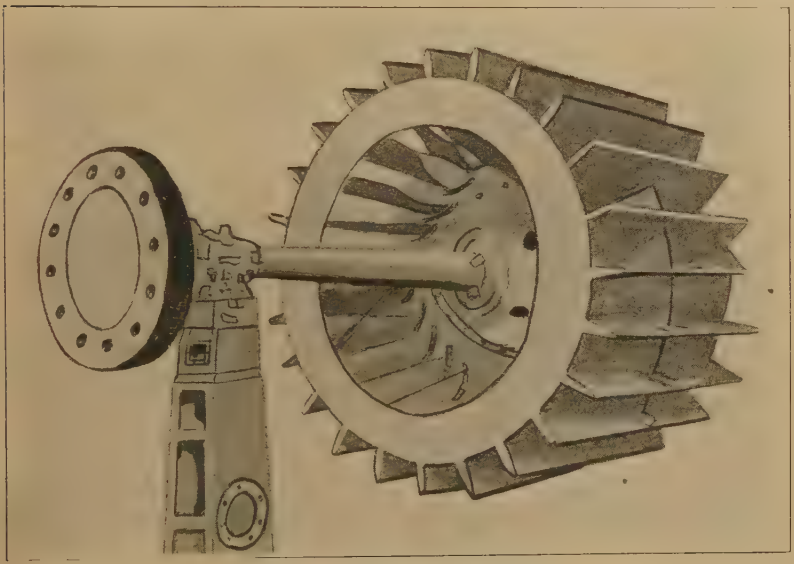


FIG. 168.—Chandler mine fan.

to all intents and purposes a shallow straight radial one, or paddle (consult figs. 80–82). A noticeable feature of the Chandler ventilator is that the volute chamber (which terminates in the evasée in all enclosed fans) is very large.

(ii) **Straight or Radially Bladed Fans.**—Fans having blades radiating like spokes without inclination or curvature, either backward or forward, are not used in this country for mine ventilation. They are chiefly used in screen-dust collection plants, and very high-speed forcing units, as in cupola work.

(iii) **Forward-bladed Fans.**—All the fans of this class at present in use in British mines are very shallow-bladed, so that the inlet is usually over 80 per cent. of the fan side area. They also have a large number of blades, sometimes in excess of sixty, high speeds up to and over 300 r.p.m., and small diameters.

The Sirocco Fan.—This important fan is much used both in this country and the Colonies. Its blades are very short radially, about 9 in., and rather longer axially than the fan width. They are bent forward in the direction of rotation and slightly cupped at the ends. The outer edges are a little narrower than the inner ones in order to avoid eddying and re-entry of air. The number of blades is 64 in the single-inlet type, and in the double-inlet type, which is simply a pair of singles arranged back to back on the same shaft, the number is 128. The blades are open at the ends toward the inflowing air. At the back end of the single, or in the middle of the double-inlet fan, is the vertical diaphragm, which has a stream-line curve to the shaft boss. This curve guides the air at the inner end directly on to the blades, so preventing rebound and banking. The rings which hold the blades are secured to the boss by six light steel tie-rods. The whole design is such that the air leaves the fan at a velocity of about 80 per cent. greater than that of a radially bladed fan of the same tip speed, consequently this fan produces as high if not a higher water gauge than any other fan, and the actual water gauge is in excess of the theoretical. The effect of this high pressure is to increase the reduction effect within the fan.

The effect of the high discharge speed is to enable the fan to be run more slowly. The Sirocco, see fig. 89, is made in sizes from 10 in. to 200 in. diameter, and the average speeds are from 100 to 280 r.p.m., the higher speeds being well suited to direct drive. It must here be noted that in a belt-drive of very low slip or creep, or in a chain-drive suitably designed, the advantage of direct drives are greatly lessened by comparison, since there is small loss; in such cases if our fan has to work at a high speed there is very small reason why our engine should also run at that speed, so when thinking of direct drives we must also make a careful comparison. The following figures supplied by the makers give a fair notion of the sizes and duties:—

Diam. in inches.	W.G. in inches.	Quantity in cub. ft. per min.	Colliery.
154 double inlet . .	12½	500,000	Carlton Main, Yorks.
140 „ „ . .	4	375,000	Llwynpia, Glam., S.W.
134 single inlet . .	2½	200,000	Harton, Durham.
98 double inlet . .	3	250,000	Hamilton, N.B.

The manometrical efficiency of the Sirocco fan is about 76 per cent.

Large diameters are not entirely advantageous, because the mechanical stresses increase with the diameter, and also in order to retain the benefit of sharing the work over a large number of blades, the blading of such fans would be prohibitive in cost. Small diameters have the drawback of requiring high revolutionary speed to enable them to attain the peripheral speed of large fans. From this and the preceding remarks it will be seen that if we wish to combine the mechanical and upkeep advantages of small-diameter fans, and at the same time have a suitable peripheral speed, the problem lies almost entirely in the design of the blade tips and approach passages.

The Keith-Blackman Fan.—This deservedly popular ventilator, though really of the sinuously bladed type, may be best classified as a forward-bladed one, since the inlet and outlet edges of the blades are curved forward in the direction of rotation. Fig. 169 shows a runner of 11 ft. over all and 10 ft. inlet diameters. There are fifty blades on each side of the centre in the double-inlet type. Each blade is riveted by a long flange to a heavy steel centre-plate, while the inlet ends are riveted to a narrow shroud-ring, strengthened by an angle-iron spigot. It is claimed that the curving forward of the blades at the inner edge enables the air to enter with the minimum of shock. The centre, by which the runner is secured to the shaft, comprises two conical iron castings, which are introduced from either side. The external diameter of the runner, over the blade tips, diminishes from the inlets to the centre. The effect of the enlargement of the outer diameter of the wheel will be apparent when it is borne in mind that the inductive power depends mainly on the velocity of the blade tips, and therefore the induction will be greatest near the inlet where the peripheral velocity is greater, and least at the back, so that the natural tendency for the air to flow to the back is exactly counterbalanced by the increased induction towards the front. Hence there is no overcrowding of the air at the back of the drum, but the whole area does equal work, and so more air is dealt with than by a parallel drum, whilst power is saved because no portion of the air is raised to a higher velocity than any other portion, as is the case where the distribution effect is unequal.

In the latter case, where air leaves a fan at various velocities the resultant pressure can only be that due to the mean of these

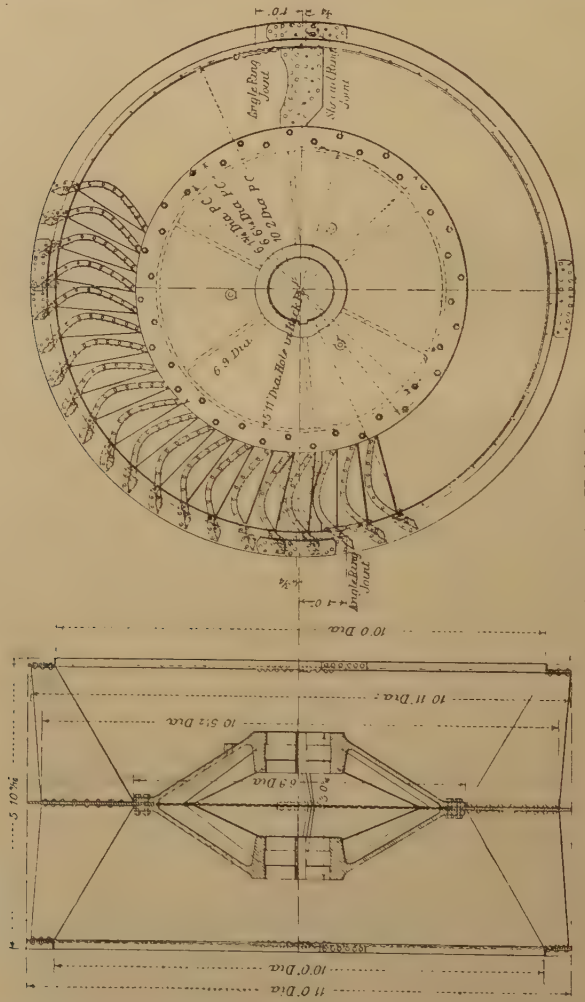


Fig. 169.—Keith fan.

velocities, and it therefore follows that part of the air must needs be raised to a higher velocity than would otherwise be necessary for the pressure obtained, and hence much power is lost by such fans because the power absorbed in moving air increases as the

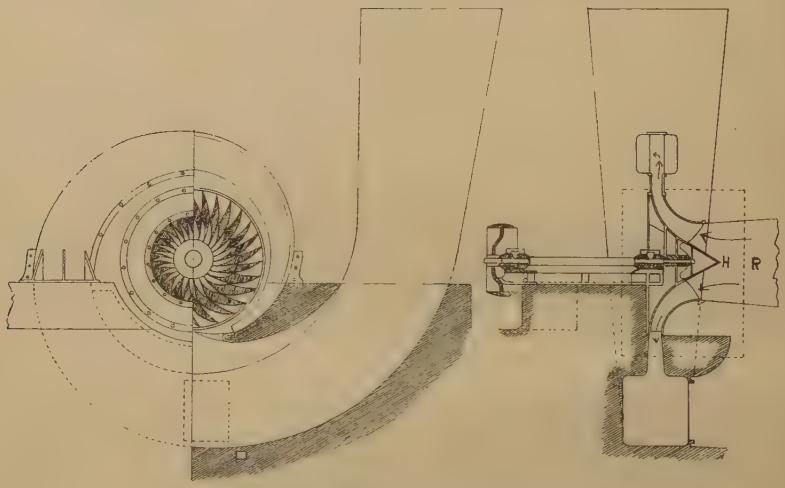
third power of the velocity (see law, *r*2), *i.e.* double the velocity means eight times the power.

Another great feature of the Keith-Blackman fan is that the blades are not stayed or gusseted. The nearly triangular shape of the blades gives them strength sufficient to resist all the centrifugal stresses, while the curvature provides ample lateral stiffness. This obviates the need for supplementary internal stays, which cause peripheral eddies in other runners of this type. The Keith fans may be fitted with pulley for belt- or rope-drive, or they can be arranged for direct steam-, electric-, gas-, or oil-engines. The manometrical efficiency is between 70 and 80 per cent.

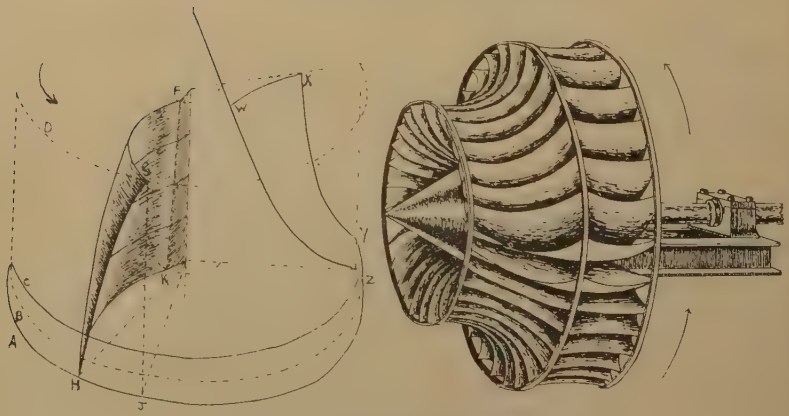
> It should be noted here that all drum-type fans with shallow blades are almost useless for pressure or suction above the few inches of water gauge common to mine ventilation. Another great fault of all parallel drum fans is the above-mentioned unequal distribution effect.

The Rateau Fan.—This fan is better known on the Continent than in this country, but its careful design, which takes into full account the performance of a particle of air in circular motion, warrants its inclusion at this point. It consists essentially of the turbine, or blading system, and the diffuser. It will be seen by figs. 170–173 that the vanes are designed in such a manner as to give sufficient lateral strength without extra staying, and also to enable the air to have an easy path through the machine. The diffuser, which is a spiral chamber of increasing rectangular section about the blade tips, is adopted in order to increase the energy-conversion effect previous to admitting the air to the volute which fits on the outer side of it and terminates in the conical evasée chimney. The general design will be better comprehended by referring to fig. 171. H is the cone-capped wheel-boss carrying the blades and designed to give a smooth stream-line effect. The blades, which number twenty-four to thirty, run with the minimum of clearance from the casing on the drive side. On the inlet side the casing is designed to give a slight increase in velocity to the air passing through the fan. The inlet approach, R, is coned toward the eye, as in the modern Waddle, so giving a uniform velocity to the inflowing air over its whole length. Only single-inlet types are made, and the blades are stamp or core-moulded in cast iron in such a sweep that all the air particles will have a similar trajectory. Fig. 172 shows the geometrical

principles involved in the design of a Rateau blade.* Imagine the centre line to be the shaft axis, and the circles through A, B, C, and D to have radii equal to the external radius of the



FIGS. 170 and 171.—Rateau fan (principle).



FIGS. 172 and 173.—Rateau fan (design).

fan. Take a point, G, on the circle D, and through it draw the arcs G F and G H. The angle at G between the arc G F and the tangent at G is usually about 45 degrees. The angle H G J

* See also C. H. Innes, *The Fan*, p. 51.

between the circular arc G H and the line G J parallel to the shaft axis is such that

$$\tan \text{HGJ} = \frac{\text{Velocity of the tips}}{\text{Normal velocity of the air}}.$$

By this means a particle of air at any point on the arc G F will meet the edge of the blade when the fan is running under the conditions for which it was designed. The points F, G, H, and K on the blade will describe the circles passing through them when the wheel revolves. The direction of revolution is shown by the arrow. The whole blade F G H K is not required, and is cut down by revolving the portion W X Y Z about the main

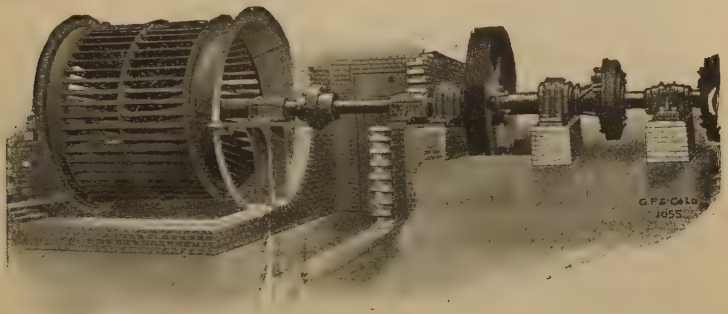


Fig. 174.—The Brightside fan.

axis so as to cut out its shape, which will be the blade. W X will be the inlet edge, and Y Z the tip edge. Fig. 173 shows in perspective the appearance of the assembled blades upon the boss.

The Brightside Fan.—Among the modern ventilating machines this reliable fan is firmly establishing itself. Its essential features, apart from long axial and forward-curved blades, typical of most modern ventilators. are mechanical ones, as will be seen by fig. 174.

It is well bourne, adaptable to direct or geared drive, and has a heavy inlet wall-casting fitting near the fan proper at each inlet. The construction throughout is massive, so overcoming some of the defects common to this kind of fan, which run at high speeds and water gauges. It is built in sizes from 70 in. diameter up to 150 in., the larger sizes being capable of dealing with over half a million cubic feet of air per minute against a 6-in. W.G.

It cannot be emphasised too often that a 6-in. W.G. will be a

common pressure as mining at great depths, especially in thin seams, becomes more prevalent. This will mean that in order to pass the above quantity per minute upwards of 800 b.h.p. will, in most cases, have to be expended. In such cases only comparatively small and very rigid multi-blade fans (but having a total weight of over a hundred tons) will meet the conditions. The Brightside fan, which has over sixty blades on either side of the central diaphragm, to which, and the outer annulus rings, they

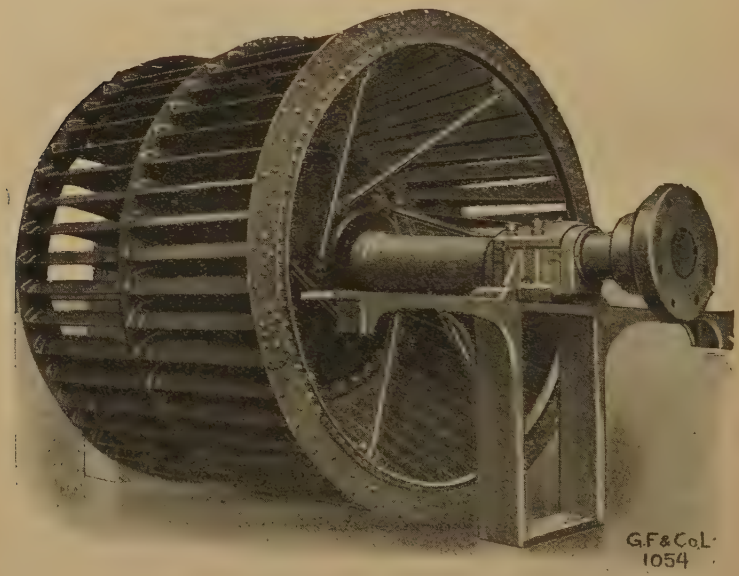


Fig. 175.—Runner of the Brightside fan.

are securely riveted, is further strengthened by six or eight tie-rods between the said rings and diaphragm boss, and by this means is well adapted to the conditions mentioned above.

(iv) **Compound-bladed Fans.**—By compound-bladed ventilators we mean those machines in which the blades are sinuous (curved both backward and forward), and also those in which the blades alternate either in shape or size.

The Turbon Fan.—This fan is constructed by the Waddle Fan and Engineering Co., Llanelly, S.W., and differs very materially from other fans. Instead of the runner being made up of a large number of shallow blades of great axial length, a few steel

rings pressed by dies and made to interlock in each other form the chief part of the runner.

Figs. 176 and 177 will make the design of the runner clear. The corrugated rings are secured between the driving disc and holding rings by means of stay-bolts, which bind the whole firmly together, adjoining rings locking their neighbours. The fact that all the rings are pressed in the same die ensures self-centering and self-balancing, and the corrugations add greatly to the torsional strength. The makers claim that the machine is silent and easily dismantled, and that the rings and casing can be reversed for counter-clockwise running—a fact which does not

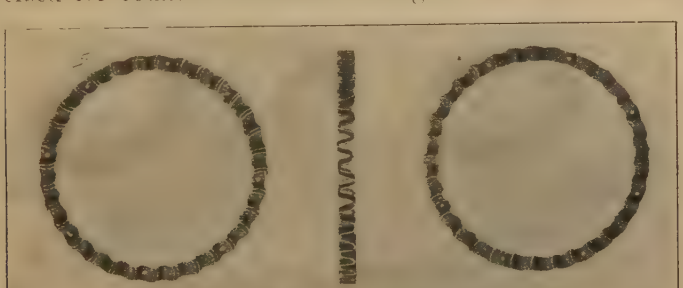


FIG. 176.—Turbon fan. Pressed-ring element.

apply to other fans. They also claim that the cellular construction given to the air passages results in the air being taken hold of more effectively than by the series of inter-blade oblong passages of other types of fans. (This in some measure accounts for the wavy form given to the blades in some of the newer fans.) From the standpoints of capacity, useful effect, mechanical principle, and the maintenance of a wide range of efficiency, this fan stands well to the fore. Turbon fans are made in sizes from 16 in. diam. up to 100 in., when the average speed is about 300 r.p.m. against 6 in. W.G. The larger sizes of the double-inlet type are capable of dealing with quantities up to 600,000 cub. ft. per minute, taking from 250 to 800 h.p. The manometrical efficiency is about 75 per cent.

The Ordnance fan also belongs to this type, but has cupped corrugations.

The Jeffrey Fan.—This is the most widely known of the American ventilators, and is certainly a very efficient machine. The fan is constructed in four distinct types, *i.e.* Blowing, Exhausting, Primarily Blowing Reversible, and Primarily

Exhausting Reversible. The blowers take the air axially direct from the atmosphere, and have their blades inclined forward in the direction of rotation, whilst the exhausters have the blades

Fig. 177.—Turbon fan. Double inlet runner with wall-box casting for building into the mouth of the fan drift.

curved back from the direction of rotation. It will be seen that the same construction is suitable for all the above four types. The blades are set in alternate sizes—short, medium, short, and then long—around the inner side of the outer edge of the wheel.

The short blades are about one-tenth the fan diameter in depth at the inner or diaphragm end, and the long ones a little over three times as deep as the short ones at the diaphragm. This stepping of the blades adds the pressure effect of the deeper blades to the volumetric effect of the shorter ones, which are

FIG. 178.—Jeffrey fan (single inlet).

set in pairs. The fan is usually of the double-inlet type, the blades being all of the same external axial length but declining to about one-third this length at the inner edge near the diaphragm. The design is similar on the opposite side of the diaphragm for double-inlet fans. The resulting shape of a blade will, therefore, be that of a bent trapezoid (see figs. 178 and 179).

The outer-inlet annulus, which holds the outer edges of the blades, has a diameter of about 80 per cent. of the fan diameter, and is braced by light steel rods to a narrow boss ring near the bearing. The whole fan and casing is made of steel throughout, and is reversible. The makers claim mechanical and manometrical efficiencies of about 80 per cent., and volume ratios up to 420. Like most modern fans, the Jeffrey is made in vary-

ing sizes, with capacities ranging up to about 500,000 cub. ft. per minute.

A common feature of step-bladed fans is that they are seldom more in width than 50 per cent. of the diameter, and in keeping with current practice are small, about 16 ft. being the maximum diameter.

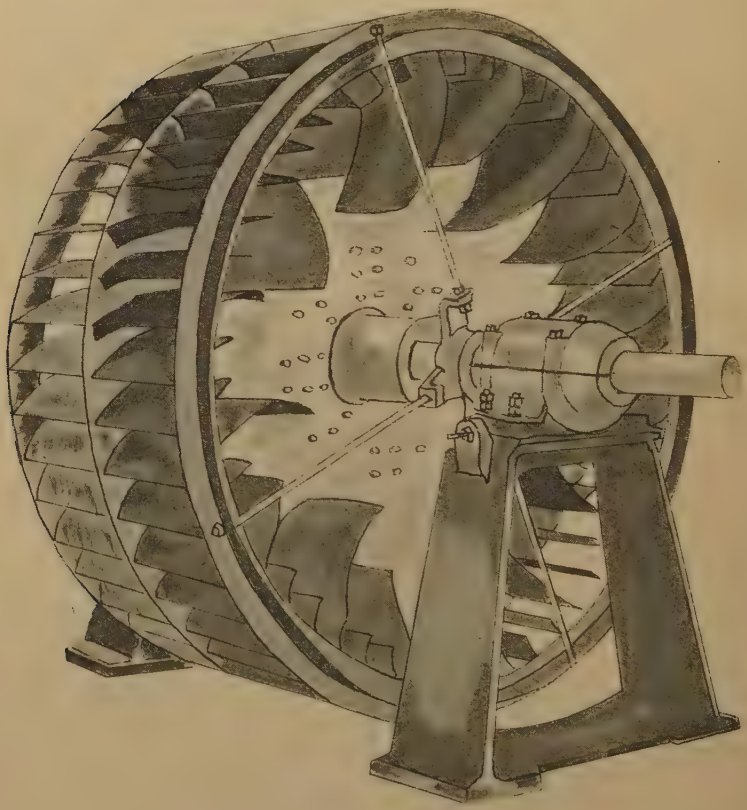


FIG. 179.—Double inlet type fan (Jeffrey).

Siemens-Schuckert Fan.—This fan (fig. 180) belongs to the alternate-bladed type, like the one previously described, the only difference being that the blades are cut straight from the outlet to the deep or diaphragm end. It is usually made in small units and installed in combinations for mass work, and for

this reason is not adopted for large mine work. Fig. 180 shows the alternate rim and boss blades of one of a complement of small fans of this type taken from a captured German submarine and now installed at the College, Loughborough. Note that the shallow or rim blades are almost entirely rectangular, while the

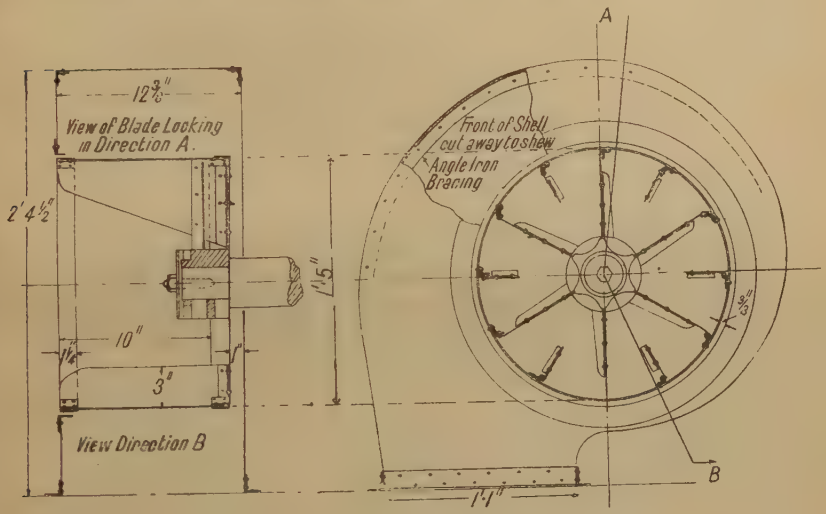


FIG. 180.—Siemens-Schuckert (Berlin.)

deeper ones are nearly triangular. (This machine is not common in this country, and is included here in order to further illustrate "step" blading.)

The Barclay Fan.—This machine, the product of the famous Kilmarnock firm, has its blades arranged in alternating step sizes about the rim, as in the two previous types discussed, the blades being short axially and the whole structure of steel throughout. The chief feature of the blading is that the deeper blades are bent in the direction of rotation into an angle of about 30 degrees forward, and increase in radial depth toward the diaphragm. A light octagonal spider braces the inlet ring, which is 75 per cent. of the fan diameter (see fig. 181). The whole design is that of strength and reliability. The sizes are rarely more than 20 ft. diameter, and the speeds vary up to 200 r.p.m., the drive being by belt or rope gearing. The maximum duties are from 400,000 to 500,000 cub. ft. per minute against pressures

of up to 5 or 6 in. of W.G. This is the type fan made by this firm, but they also produce a shallow-bladed non-stepped fan for similar duties.

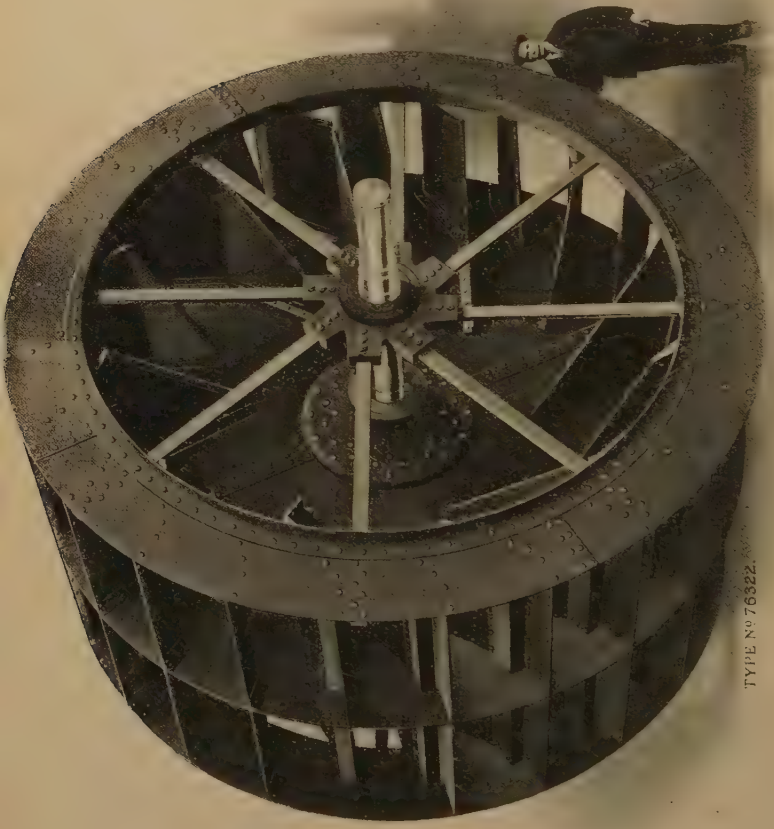


FIG. 181.—"Barclay" fan runner.

The Capell Fan.—This important ventilator was formerly made with only from twelve to sixteen blades, each of which radiated from the boss, the first half of its length being bent forward in the direction of rotation so giving a scooping effect to the air

on its entry to the vanes, and the outer half bent backward from the direction of rotation aiding the discharging effect. The modern type has double the number of blades, *i.e.* thirty-two.

The extra blades found in the modern Capell are in the nature of wings placed in varying positions near the outer edge between the blades proper. Their purpose is to modify the radial velocity of the air passing through the fan. The modern Capell (see figs. 182 and 183) has a projecting guide edge on its inlet

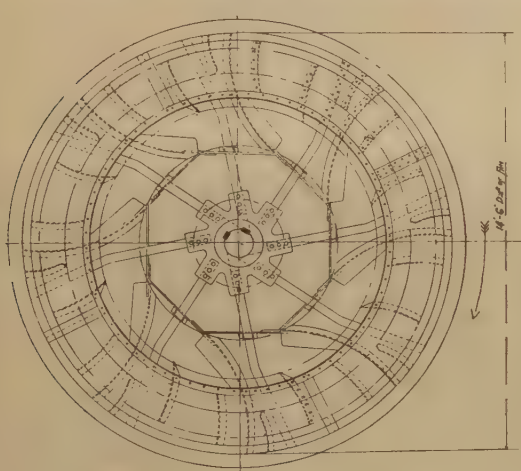


Fig. 182.—Elevation of double-inlet Capell fan, 14½ ft. by 5 ft. 10 in.

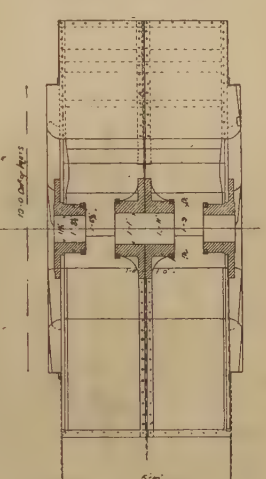


Fig. 183.—Section of double-inlet Capell fan, 14½ ft. by 5 ft. 10 in.

sides which conducts the air smoothly on to the vanes. Such a fan will deal with 350,000 cub. ft. of air per minute against a 6-in. W.G. when running at 200 r.p.m., being 70 per cent. efficient, and having (when electrically driven) an overall efficiency of about 60 per cent. Since the above figures are taken from actual practice * the modern Capell fan must certainly be placed among the best ventilating machines in practice at present.

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CHAPTER X

THE THEORY OF THE MINE FAN

We purpose giving in this chapter a few of the more popular aspects of the main principles underlying the design of fans, the passage of air through the fan, and other fundamental details.

Loss of Head due to Change in Direction of the Air.—By Fig. 66 (which is reproduced in fig. 184) and formula (129), it was shown that the loss of head due to a sudden change in the direction of the air is given by

$$\frac{BC^2}{2g} = \frac{V_1^2 + V_2^2 - 2V_1V_2\cos\alpha}{2g} \quad . \quad . \quad (222)$$

where V_1 is the former velocity and V_2 the velocity after the direction has been changed through an angle α. This expression

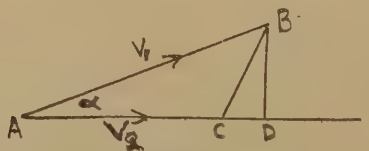


FIG. 184.

will have its least value when B C and the perpendicular B D coincide, then

$$V_2 = V_1\cos\alpha \quad . \quad . \quad . \quad . \quad (223)$$

Substituting (223) in (222), the *minimum loss of head* will be

$$\frac{V_1^2 - V_1^2\cos^2\alpha}{2g} = \frac{V_1^2(1-\cos^2\alpha)}{2g} = \frac{V_1^2\sin^2\alpha}{2g} \quad . \quad (224)$$

Conditions for Gain of Head.—Since the pressures met with in mines are seldom over 10 in. of water column, which is not 6 oz. per square foot, and in the majority of cases is only about a third

of this, it introduces no appreciable error to assume that the density of the air remains unchanged throughout the fan.

If the pressure corresponding to the velocity V_1 be p_1 and that after the change be p_2, and the density d, then by (63)

$$\frac{p_1}{d}+\frac{V_1^2}{2g}-\text{loss of head}=\frac{p_2}{d}+\frac{V_2^2}{2g} \quad . \quad . \quad . \quad (225)$$

$$\frac{p_2-p_1}{d}=\text{gain of head}=\frac{V_1^2-V_2^2}{2g}-\text{loss of head}$$

$\therefore$ by (222) gain of head $= \dfrac{V_1^2-V_2^2}{2g}-\left(\dfrac{V_1^2+V_2^2-2V_1V_2\cos\alpha}{2g}\right)$

$$=\frac{V_1V_2\cos\alpha-V_2^2}{g} \quad . \quad . \quad . \quad . \quad (226)$$

differentiating and equating to zero for maxima

$$\frac{\delta}{\delta V_2}(V_1V_2\cos\alpha-V_2^2)=0$$

$$=V_1\cos\alpha-2V_2=0$$

$$\therefore \quad V_2=\frac{V_1\cos\alpha}{2} \quad . \quad . \quad . \quad . \quad (227)$$

i.e. maximum gain of pressure takes place when

$$AC=\frac{AD}{2}=CD.$$

From this it will be seen that where there is a sudden change in direction, as in fans, and an evasée is supplied to recoup as much pressure head as possible, the minimum loss will be obtained when equation (224) is satisfied; and where there is no evasée the maximum gain of pressure head will occur when the design is in accordance with (227)—see notes on the Waddle fan.

(The above procedure is adopted in a similar manner and to a much fuller extent in C. H. Innes' splendid little work, *The Fan*, a book which, though it does not deal with mine ventilation, should be possessed by everyone anxious to pursue the subject of fan design *in extenso*.)

Gain of Pressure due to Fall in Velocity.—If the velocity falls from V_1 when the pressure is p_1, to V_2 without changing its direction, the pressure will increase to, say, p_2 in accordance with the law of the conservation of energy.

The loss of head will be

$$\frac{(V_1 - V_2)^2}{2g} \quad . \quad . \quad . \quad . \quad (228)$$

and the gain in pressure head if d is the density will be

$$\frac{p_2 - p_1}{d} \quad . \quad . \quad . \quad . \quad (229)$$

as by the method for (226), therefore

$$\frac{p_2 - p_1}{d} = \frac{V_1^2 - V_2^2}{2g} - \frac{(V_1 - V_2)^2}{2g}$$

$$= \frac{V_1 V_2 - V_2^2}{g} \quad . \quad . \quad . \quad . \quad (230)$$

where V_1 is the evasée entrance velocity and V_2 that of the exit.

The Acceleration of the Air through the Fan.—In fig. 185 the inner parallelogram of velocities is denoted by small lettering and the outer one by large letters in order to facilitate the grouping of similar effects.

pP is a blade bent backward from the direction of rotation, which is clockwise; r and R are its respective inner and outer radii. The air entering the fan axially is turned radially on the centre line so that a particle will enter the blades with a radial velocity of v_1 feet per second, represented to scale by pm. The inner circumferential velocity v is shown to scale as the tangent po, which with the velocity a (representing the backward velocity of the air over the blade shown by the tangent pl) form the components of the inner radial velocity, v_1. All the components and therefore resultants being to scale from direct running data, it will be seen that $v = v_1 \cot i$, the angle i being that between the tangent to the inner end of the blade and a tangent to the circumference at p.

Also $$a^2 = v^2 + v_1^2 \quad . \quad . \quad . \quad . \quad (231)$$

The fan wheel will accelerate the air in its passage through to the blade tip P, but in the meantime P has travelled to P′, and the absolute path of the air will be that shown by the dotted line p, P′, as by the method of fig. 75. The air is delivered into the volute with an absolute velocity, V_2, tangential to the absolute path at the tips and of a magnitude compounded of the peripheral velocity of the wheel tip, *i.e.* V, and the velocity of the air relative to the blade at the tip shown by the blade tip tangent PL = A,

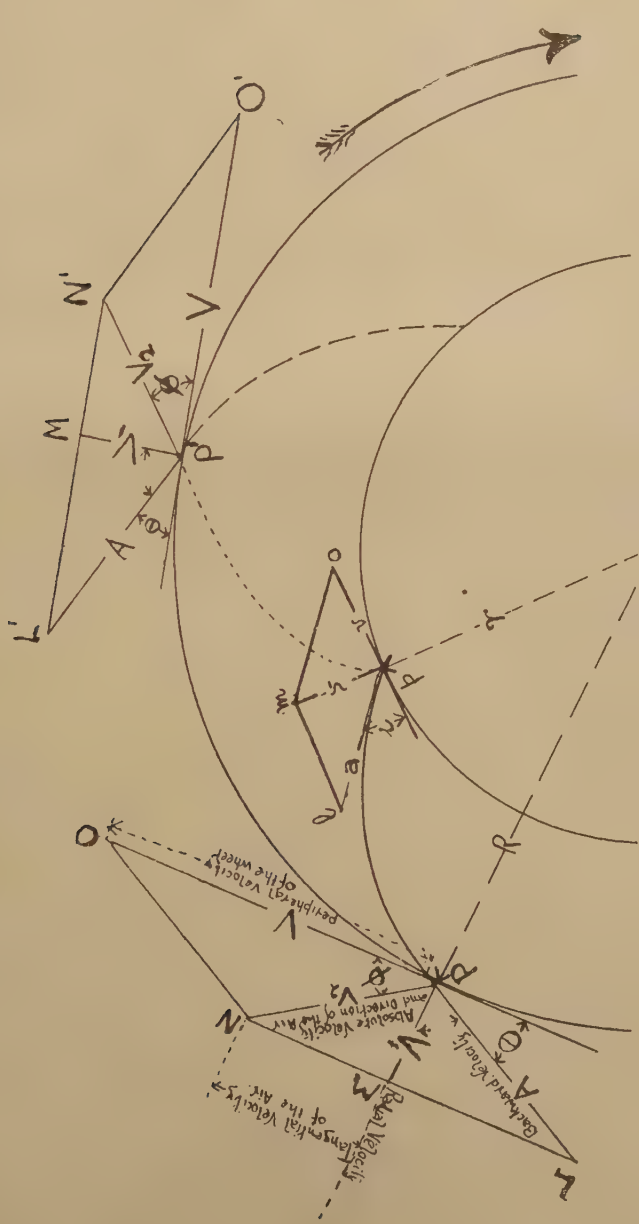

Peripheral Velocity of the wheel
Backward Velocity
Absolute Velocity and direction of the Air
Tangential Velocity of the Air

FIG. 185.—Vectorial diagram for backward-bladed fan.

to scale like all the other velocities. The air leaves the runner at an angle ϕ, and consequently the relative velocity PL is

$$A^2 = V^2 + V_2^2 - 2VV_2 \cos \phi \quad . \quad . \quad (232)$$

If h_1 is the pressure head required to cause the entry velocity v_1 and f is the coefficient of resistance at entry, then $h_1 = \frac{v_1^2}{2g} + f\frac{v_1^2}{2g}$, and this is the difference between the absolute pressure in the drift and that at the point p. Let h be the static mine pressure in the drift and B the atmosphere pressure, then the total absolute pressure b, at p, is

$$b = B - h - h_1 \quad . \quad . \quad . \quad . \quad (233)$$

The static difference between the atmosphere and the air at the end of the evasée is nil, but the velocity here is, say, w feet per second, and if f_1 is the coefficient of resistance at this point then the air leaving the runner at P′ with the absolute velocity V_2 must possess an absolute static pressure, h_2. This pressure, h_2, is greater than $\left(B + f_1\frac{w^2}{2g}\right)$, since it also sustains the exit velocity w, therefore

$$h_2 + \frac{V_2^2}{2g} = B + f_1\frac{w^2}{2g} + \frac{w^2}{2g} \quad . \quad . \quad . \quad (234)$$

The energy imparted to the air by the fan from the inlet point p to the exit point P′ is equal to the difference of the pressure heads represented by the peripheral velocities v and V at these points respectively, thus—

$$h_2 + \frac{A^2}{2g} - \left(b + \frac{a^2}{2g}\right) + f_2\frac{V_1^2}{2g} = \frac{V^2 - v^2}{2g} \quad . \quad . \quad (235)$$

where f_2 is the coefficient of resistance set up by the blades upon the air in its passage through the fan.

Substituting in (235) the values of a, A, b, and h_2 from the preceding four formulæ and clearing the expression, we get—

$$h + f\frac{v_1^2}{2g} + f_1\frac{w^2}{2g} + f_2\frac{V_1^2}{2g} + \frac{w^2}{2g} = \frac{2VV_2 \cos \phi}{2g} \quad . \quad . \quad (236)$$

and calling the sum of the friction terms, h_0, this becomes

$$h + h_0 + \frac{w^2}{2g} = \frac{2VV_2 \cos \phi}{2g} \quad . \quad . \quad (237)$$

or $$g(h+h_0)+\frac{w^2}{2}=VV_2\cos\phi \quad . \quad . \quad . \quad (238)$$

By fig. 185 it will be seen that

$$\frac{V}{V_2}=\frac{\sin(180-(\phi+\theta))}{\sin\theta}$$

whence $$V=V_2\frac{\sin(\phi+\theta)}{\sin\theta} \quad . \quad . \quad . \quad . \quad (239)$$

Substituting V_2 from (239) in (238)

$$V^2=\left[\frac{g(h+h_0)+\frac{w^2}{2}}{\cos\phi}\right]\frac{\sin(\phi+\theta)}{\sin\theta}$$

whence $$V=\sqrt{\left[g(h+h_0)+\frac{w^2}{2}\right]\frac{\sin(\phi+\theta)}{\sin\theta\cos\phi}} \quad . \quad . \quad (240)$$

This gives the peripheral velocity when the pressure h, the total fan resistance h_0, and the discharge velocity w are known, also the angles θ and ϕ. The angle θ is got directly, being the backward curvature of the blades at the tips, and ϕ is easily obtained.

With radial blades θ is 90° and the above expression becomes

$$V=\sqrt{g(h+h_0)+\frac{w^2}{2}} \quad . \quad . \quad . \quad (241)$$

The peripheral velocity can, of course, be expressed in various ways according to the terms employed.

Formula (241) above is really a modified form of (101), which can be shown by clearing the radical, when

$$V^2=g(h+h_0)+\frac{w^2}{2} \quad . \quad . \quad . \quad (242)$$

The total pressure of the air, which we have previously denoted by H_1, is here made up of the static head h, the head to overcome the fan resistance h_0, and that due to the velocity w, wherefore (242) may be written

$$\frac{V^2}{g}=h+h_0+\frac{w^2}{2g}=H_1 \quad . \quad . \quad . \quad (243)$$

and therefore (101) $$H_1=\frac{V^2}{g}.$$

The Effect of Blade Curvature.—*Backward Bent Blades.*—The air passing out of the fan moves over the blades at such a rate

that its radial component relative to the blade is V_1, its actual component relative to the blade being A. The velocity V_2, with which the air leaves the blades, is the resultant of the peripheral velocity of the wheel, *i.e.* PO = V, and the relative or backward component PL = A, so that PN represents V_2 in magnitude and

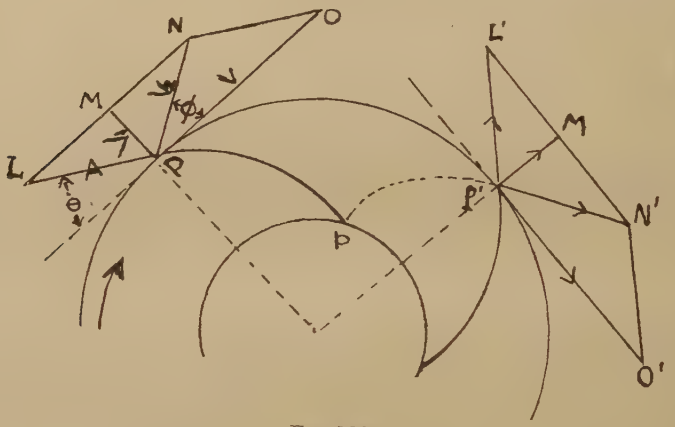


FIG. 186.

direction. The angle made by the blade tips with the circumferential tangent is θ, therefore $LM = V_1 \cot \theta$ and $LN = PO = V$.

$$\therefore \text{ MN (the tangential velocity of the air)} = LN - LM = V - V_1 \cot \theta \quad . \quad . \quad . \quad (244)$$

By Newton's Second Law the momentum of every pound of air leaving the fan is $\frac{V - V_1 \cot \theta}{g}$, and every pound of air leaving the fan per second exerts a retarding force on the fan equal to this amount (see the Third Law of Newton). The fan overcomes this force at a rate of V feet per second, so that the work done, which by Definitions is equivalent to force times distance, is

$$\frac{V^2 - VV_1 \cot \theta}{g} \text{ ft.-lbs.}$$

This is the height of the column of air which the fan will support.

$$\therefore H_1 = \frac{V^2 - VV_1 \cot \theta}{g} \quad . \quad . \quad . \quad (245)$$

Radial Blades.—If the blades are straight radial ones θ will be 90° and the above expression becomes

$$H_1 = \frac{V^2}{g},$$

the same as that of (101), consequently the formulæ of the previous chapters can only apply to straight, radially bladed fans.

Forward Bent Blades.—The accompanying figure preserves the above reasoning, but applied to a forward-bladed fan. Here

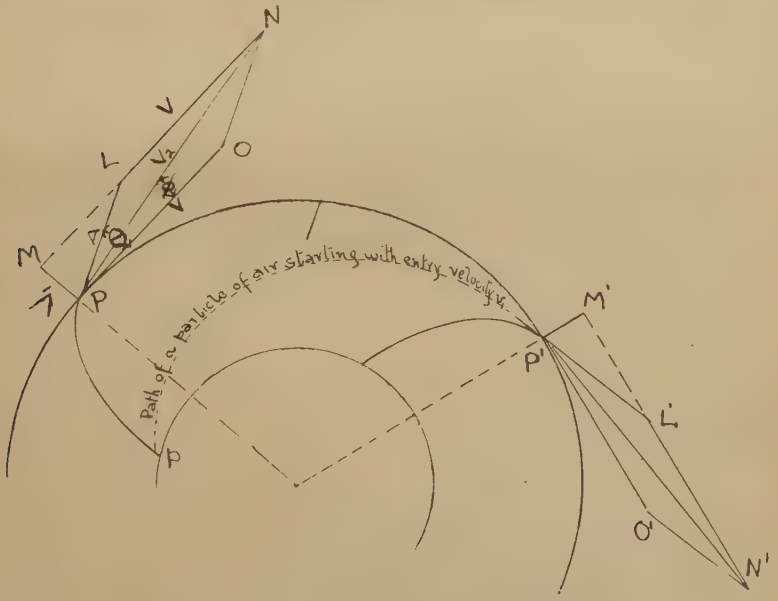


FIG. 187.—Vectors of a forward-bladed fan.

the tangential component of the resultant velocity is greater than V by $V_1 \cot \theta$, so that $MN = V + V_1 \cot \theta$, and by the same method as before the theoretical head is

$$H_1 = \frac{V^2 + VV_1 \cot \theta}{g} \quad . \quad . \quad . \quad (246)$$

A general expression for the theoretical head for both backward and forward bladed fans will be therefore

$$H_1 = \frac{V^2 \pm VV_1 \cot \theta}{g} \quad . \quad . \quad . \quad (247)$$

The positive sign applies to the forward and the negative to the backward bladed fan.

By (102) this may be expressed in inches of water gauge thus—

$$\frac{12d(V^2 \pm VV_1 \cot \theta)}{d_1 g} \text{ in. W.G. (max.)} \quad . \quad . \quad (248)$$

Dividing this into the mine W.G., the manometrical or true mechanical efficiency is obtained thus for any of the above types of fan—

$$\frac{\text{W.G. } d_1 g}{12d(V^2 \pm VV_1 \cot \theta)} \quad . \quad . \quad . \quad (249)$$

It will now be manifest that to apply (101) to all fans is erroneous, it being only correct for radially bladed fans. To get a correct calculation for the particular fan one is dealing with (249) above must be applied with the sign appropriate to the type. This also emphasises our remarks with respect to the old equivalent orifice methods in a previous chapter. The increase in efficiency of backward bent blades over radial ones is seen to exist in the inclusion of the term, $-VV_1 \cot \theta$, in the denominator, thus giving a greater ratio and destroying the anomalies mentioned in the said previous chapter.

The radial velocity, V_1, can be obtained by considering the quantity, Q, dealt with per second and the internal and external axial widths of the blades l_1 and l_2 respectively. By fig. 185 it will be clear that $V_1 = V_2 \sin \phi$, and

$$Q = 2\pi R l_2 V_2 \sin \phi = 2\pi r l_1 v_1 \quad . \quad . \quad (250)$$

$$\therefore \; v_1 = \frac{R l_2}{r l_1} V_2 \sin \phi \quad \text{and} \quad \tan i = \frac{v_1}{v}.$$

But usually $l_1 - l_2$, so that this *radial velocity of entry* becomes

$$v_1 = \frac{R}{r} V_2 \sin \phi.$$

The radial velocity of exit which is the most important here is

$$V_1 = \frac{Q}{2\pi r l_2} \quad . \quad . \quad . \quad . \quad (251)$$

The radial velocity at any point in the fan can, in a similar way, be shown to vary inversely as the fan width at that point, and also as its distance from the centre.

The manometrical efficiency, $\frac{\text{W.G.} \times d_1 \times g}{12 \times d \times (V^2 \pm VV_1 \cot \theta)}$, shows that backward-bladed fans are more efficient when the radial velocity V_1 is large, and for forward-bladed fans when V_1 is small. But (251) shows that V_1 varies inversely as the diameter and width of the fan. To increase it the fan may be decreased in diameter, or the width may be decreased as in the case of the Waddle, and to a less degree in most other types of backward-bladed fans. Forward-bladed fans must, we have seen, have a radial velocity decreased somewhat from that required by a radially bladed machine. In order to accomplish this, modern fan-makers do not increase the diameter because this would necessitate a greater expenditure of power—they increase the axial width. This, of course, means that the revolutionary speed must be raised in order to maintain the peripheral velocity, hence the reason why modern forward-bladed ventilators are small, wide, and quick-running. The width of the *double-inlet* Sirocco fan is about 20 per cent. greater than the diameter. Other fans achieve this lowered radial velocity effect by the introduction of extra and specially shaped blades or wings, as was seen in the case of the Capell.

From the foregoing remarks it will be clear that the efficiency of any ventilator depends primarily on the way in which the air enters, is conducted through the machine, discharged at the tips, and finally the path of the volute and evasée.

In radially bladed fans or those with radial tips only, the theoretical water gauge depends only on the peripheral speed ; it is not affected by the quantity delivered. Consequently, the greatest static water gauge is got when the drift is closed off, as stated elsewhere. Fans with backward-curved blades are subject to different factors, as stated in the chapter on "Mine Pressures and Resistances," section "Initial Water Gauge," for here the theoretic water gauge which they produce depends in some measure on the quantity passed by the fan. The effect of increase of quantity is to reduce the theoretical water gauge with backward-bladed fans and the reverse with forward-bladed fans. We have seen that the initial depression in the drift is not a maximum with the latter type when the drift is closed, there being a slight rise when the doors are gradually opened and then a fall. With backward-bladed fans there is no such initial rise in the gauge, the fall being continuous from the commencement of opening to full open. The larger the quantity delivered the higher

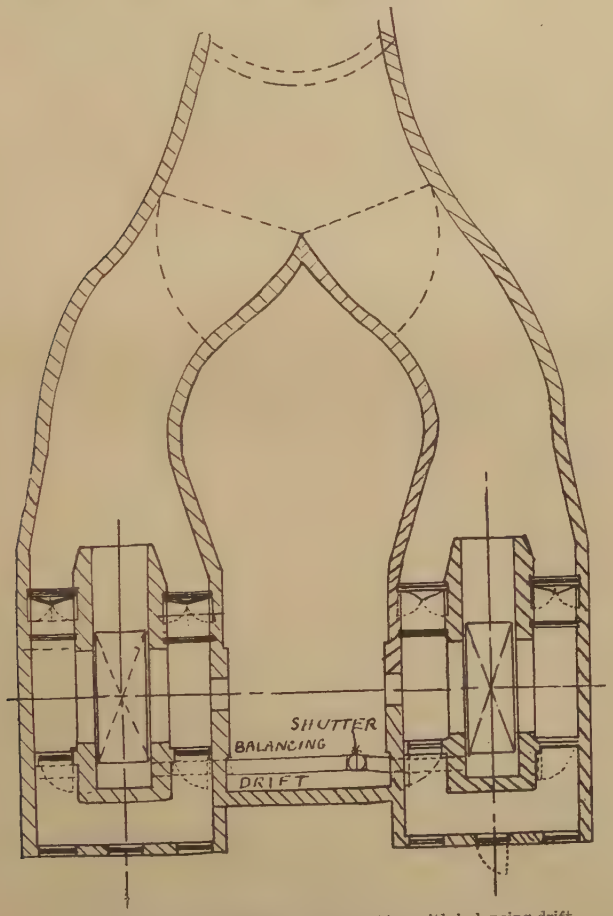
the water gauge produced with a forward-bladed fan, and also the greater the portion lost in the fan itself, and in time this fan resistance will outweigh any advantage due to forward curvature, when the static water gauge in the fan drift will start to decrease.

Combinations of Fans.—When speaking of fans operating in combination, we do not include underground auxiliaries, because their effect on the general ventilating system can usually be considered negligible. What is meant is the combined operation of large fans, either of which might be able to ventilate the mine alone for a period of time.

As in the case of centrifugal pumps, two or more ventilators can be arranged to work in series, *i.e.* in line, or in parallel, *i.e.* side by side. It will be seen that series fans will increase the initial depression for a given peripheral speed in the ratio of the number of fans, but the friction is also increased since the air has to pass through more than one fan. Again, keeping in mind the simile of the pumps, if the fans were in parallel the initial depression would not be increased, since they have both separate outlets, but the friction would be decreased owing to the quantity of air having more than one road to travel instead of being crammed through a single airway.

It is not absolutely necessary that such fans be exact replicas of each other, though such a condition is desirable, but what is important is that the manometric and volumetric capacities of each must be the same. This means that separate fans must be so driven that they maintain the same effective tangential speeds. If they were not so arranged they would have unequal effects, with the result that the stronger one would pull its air through the weaker one direct from the atmosphere when working in parallel, and when arranged in series the weaker one would have an undue throttling effect on the larger fan. The operation of fans in combination, especially in parallel, will thus need a means of balancing to prevent this "robbery," which means may take the form of a tube connecting the two fans. Such a synchronising tube is shown in fig. 188, where there are two similar fans arranged for parallel working. By means of this synchronising tube or balancing drift the orifices of each fan can be kept the same under varying conditions of speed, inlet area, etc. This command of the orifice is extremely important, because "with a fixed orifice duplicate fans in parallel increase the volume with increase of water gauge, the increase being in proportion to the orifice." "Two fans which are not duplicates will increase the volume

provided they are each capable of maintaining the required water gauge, the increase being proportionate to the power of the incoming fan and the mine conditions." *



Double inlet fans arranged for parallel working with balancing drift.

FIG. 188.—Synchronising drift for ventilators working in combination.

The analogy between electricity and mine ventilation is a very close and oft-quoted one, and will be adopted here in order the more easily to elucidate the points at issue. Those requiring a fuller and more detailed exposition of the subject, accompanied

* Mr E. Seymour Wood, *Trans. Inst. M.E.*, lxiii. 109.

by a wide collection of data, charts, and numerous diagrams, are advised to seek the main sources of our present discussion.*

The complete electrical simile exists when consideration is directed toward a direct current dynamo using the whole of its potential in overcoming the resistance of the circuit and not in driving motors. The total resistance of this machine, as in the case of ventilation, is that of the circuit R, plus that of the machine itself, r. If the E.M.F. is E volts and the terminal voltage e_1 volts, and that lost in the machine e_2 volts, the current being I amps., then

$$I = \frac{E}{R+r} = \frac{e_1}{R} = \frac{e_2}{r}.$$

This, expressed in the ventilation terms with which we are already familiar, becomes, as stated in (128), $Q^2 = \frac{H}{R+r} = \frac{h}{R} = \frac{h_0}{r}$, where R is the mine resistance and r that of the fan. Q^2 is used here instead of Q, because the pressure varies as the square of the quantity (see law $p2$). The sum effect R of a number of electrical resistances, r_1, r_2, r_3, . . . etc., in series is $R = r_1 + r_2 + r_3$ . . . etc., and in parallel $\frac{1}{R} = \frac{1}{r_1} + \frac{1}{r_2} + \frac{1}{r_3}$ . . . etc. To make the ventilation analogy complete, bearing in mind the relationship between Q and p, the above becomes for fans in parallel

$$\frac{1}{\sqrt{R}} = \frac{1}{\sqrt{r_1}} + \frac{1}{\sqrt{r_2}} + \frac{1}{\sqrt{r_3}} \text{ . . . etc.}$$

This for two fans in parallel becomes

$$\frac{1}{\sqrt{R}} = \frac{1}{\sqrt{r_1}} + \frac{1}{\sqrt{r_2}}$$

$$\therefore \quad R = \frac{r_1 r_2}{(\sqrt{r_1} + \sqrt{r_2})^2} = \frac{r_1 r_2}{r_1 + 2\sqrt{r_1 r_2} + r_2} \qquad . \quad (252)$$

and if the internal resistances r_1 and r_2 are equal

$$R = \frac{r^2}{4r} = \frac{r}{4} \qquad . \quad . \quad . \quad . \quad (253)$$

* "The Operation of Fans in Parallel," Mr Joseph Parker, *Trans. Inst. M.E.*, lxii. 251; "New Method of Measuring Ventilating Resistances," Mr D. Penman, *Trans. Inst. M.E.*, lxii. 39; "Running Fans in Parallel at Bentley," Mr R. Clive, *Trans. Inst. M.E.*, lix. 135; "Characteristic Fan Curves, etc.," Mr Joseph Parker, *Trans. Inst. M.E.*, lxiii. 222; "Experiments with Duplicate Fans, etc." Mr Seymour Wood, *Trans. Inst. M.E.*, lxiii. 93.

Fans in Series.—If there are two dynamos connected in series giving E.M.Fs. of E_1 and E_2, and having internal resistances of r_1 and r_2 respectively, the exterior resistance being R, the current, I, will be

$$I = \frac{E_1 + E_2}{R + r_1 + r_2}.$$

Assuming the same E.M.F. produced by either machine when operating singly, then when the first dynamo worked alone on the circuit the power would be

$$I_1 E_1 = \frac{E_1^2}{R + r_1},$$

and if the other worked alone it would be

$$I_2 E_2 = \frac{E^2_2}{R + r_2}.$$

Working together in series the power would be

$$I(E_1 + E_2) = \frac{(E_1 + E_2)^2}{R + r_1 + r_2}.$$

If $E_1 = E_2 = E$ and $r_1 = r_2 = r$, the above becomes $\frac{4E^2}{R + 2r}$, so that if r is small compared with R, the power is practically four times that obtained from one machine. Each dynamo engine would, of course, develop twice the power needed when driving singly. If the sum of the powers developed when running singly and when running together are equal, the output from a series combination is only that of the sum of the two machines taken separately. In order to maintain this equality of power when working in combination each motor would have to be slowed down till its E.M.F. was the square root of that when running alone, and consequently the series voltage would be $\sqrt{2}E$ and the current $\sqrt{2}I$.*

Now consider the simile of two fans of internal resistances r_1 and r_2 working on a mine of resistance R, and producing water gauges h_1 and h_2. The quantity $= Q = \sqrt{\frac{h_1 + h_2}{R + r_1 + r_2}}$, but for one fan working alone it is $q_1 = \sqrt{\frac{h_1}{R + r_1}}$ and $q_2 = \sqrt{\frac{h_2}{R + r_2}}$, and if

* Mr David Penman, *Trans. Inst. M.E.*, lxii. 42.

$h_1 = h_2 - h$, and $r_1 = r_2 = r$, then $q_1 = q_2 - q = \sqrt{\frac{h}{R+r}}$, and $Q = \sqrt{\frac{2h}{R+2r}}$. If r is very small compared with R then $q = \sqrt{\frac{h}{R}}$ and $Q = \sqrt{\frac{2h}{R}}$,* or *the quantity produced by two similar fans in series is $\sqrt{2}$ times that produced by one provided that each fan produces the same total water gauge when in combination as when operating singly.* The power required will be greater than that obtained from the sum of the single powers, as was seen in the electrical

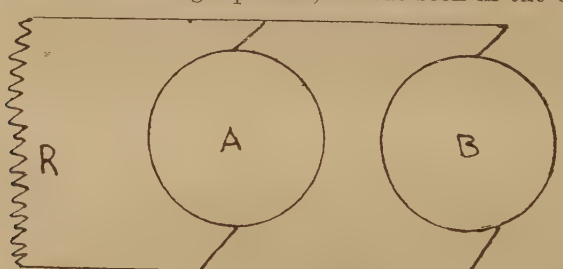


FIG. 189.

analogy. Instead of being hq for each, as given by (22), *i.e.* $2hq$ for both, the power when combined will be $\sqrt{2}$ times greater or $\sqrt{2}(2hq) = 2{\cdot}828hq$. Thus it will be seen that if we are going to maintain a *power* in combination equal to the sum of the two powers operating separately the quantity will not be increased in the proportion of the $\sqrt{2}$ as above, but as the $\sqrt[3]{2}$ (see law Q9), *i.e.* 1·26 times that produced by one fan alone. Hence the law—*the quantity produced by two similar fans in series is $\sqrt[3]{2}$ times that produced by one provided that the total power of the combination is equal to that of the sum of the fans when acting singly, and the internal resistances are comparatively small.* Note that by *similar* is meant equal in manometric and volumetric capacities and the other conditions mentioned above, and not necessarily in design, dimensions, etc.

Fans in Parallel.—Reverting again to the electrical analogy, it will be seen that when there are two dynamos (A and B, fig. 189) giving E.M.Fs. E_1 and E_2, and having internal resistances r_1 and

* Mr David Penman, *Trans. Inst. M.E.*, lxii. 43, and discussion thereon by Mr Joseph Parker, p. 60.

r_2 respectively, the external resistance being R, if $E_1 = E_2 = E$ the current is $I = \dfrac{E}{R + \dfrac{r_1 r_2}{r_1 + r_2}}$.

If $r_1 = r_2 = r$, the above becomes $I = \dfrac{E}{R + \dfrac{r}{2}}$ . . . . (254)

and each machine will take respective currents of i_1 and $i_2 = \frac{1}{2}I = \dfrac{E}{2R + r}$.

Note that as in the case of parallel centrifugal pumps the head or pressure (*i.e.* the potential) is the same with one as with two pumps, though there will be a small terminal voltage rise. When the machines have different E.M.Fs. the highest potential is also that of the circuit. When one machine operated alone the current would be $I_1 = \dfrac{E}{R + r}$, so that when the machines are paralleled there is an increase in the current circulated. This increase will be great when r is great compared with R, and negligible if r is small compared with R. The power absorbed in the circuit if both machines are identical is $IE = \dfrac{E^2}{R + \dfrac{r}{2}}$ as against $\dfrac{E^2}{R + r}$ for one alone, *i.e.* power and current would be similarly increased, showing that by paralleling the efficiency is increased over that of a single machine. When $E_1 = E_2$, if $r_1 > r_2$, i_2 will be $> i_1$; or $i_1 r_1 = i_2 r_2$. If E_1 is $< E_2$, and $r_1 = r_2$, i_1 will be $< i_2$. When $E_1 = E_2 - i_2 r_2$, $i_1 = o$, then i_2 becomes I. When E_1 is $< E_2 - i_2 r_2$, the current in r_1 is reversed. Similar results arise in ventilating with fans in parallel.*

Now since in the case of parallel ventilators

$$\frac{1}{\sqrt{R}} = \frac{1}{\sqrt{r_1}} + \frac{1}{\sqrt{r_2}} \quad \text{. . . etc.,}$$

the sum of two air resistances in parallel we have seen

$$= \sqrt{R} = \frac{\sqrt{r_1 r_2}}{\sqrt{r_1} + \sqrt{r_2}}$$

* Mr David Penman, *Trans. Inst. M.E.*, lxii. 42.

so that when $r_1 = r_2 = r$, $R = \frac{r}{4}$ as found when dealing with the electrical analogy. Then $Q = \sqrt{\frac{h}{R + \frac{r}{4}}}$ for two fans in parallel, each of equal internal resistance $= r$. In the case of one fan $Q_1 = \sqrt{\frac{h}{R + r_1}}$, r_1 being the resistance of the fan. In the case of unequal resistances $Q = \sqrt{\frac{h}{R + \frac{r_1 r_2}{(\sqrt{r_1} + \sqrt{r_2})^2}}}$ where r_1 and r_2 are the respective fan resistances.

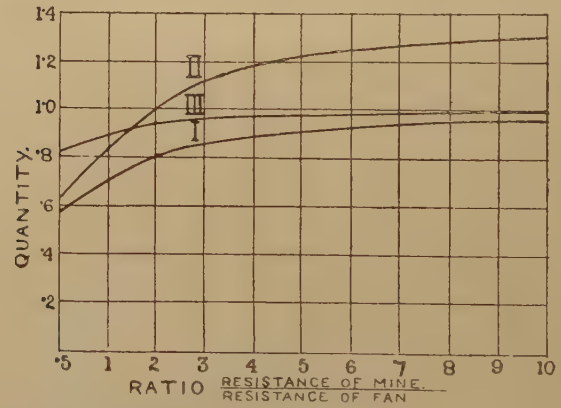


FIG. 190.—Quantity of air delivered by mine fans under different conditions: (I) single fan; (II) two identical fans in series; (III) two identical fans in parallel.

Again, as in the case of the series combination of fans, the resulting increase in quantity will depend on the conditions of combination. If a constant water gauge is maintained obviously the quantity will be different from that got when the conditions are such that the same horse-power is being maintained after as before, combining *i.e.* when either fan worked singly.

Fig. 190 is taken from the source of the above remarks due to Mr Penman, the condition being such that the water gauge pro-

duced by the fans when running together is the same as when either fan runs alone. The ordinates are in hundreds of thousands of cubic feet of air per minute, and the abscissæ the ratio $\frac{R}{r}$, from which it will be seen that from a quantity standpoint the series method of combination is the best.

By applying the Air-System Efficiency expression, the same authority constructed the chart of fig. 191, from which it will be seen that the parallel method of combination is the most efficient, the series method making for decreased efficiency.

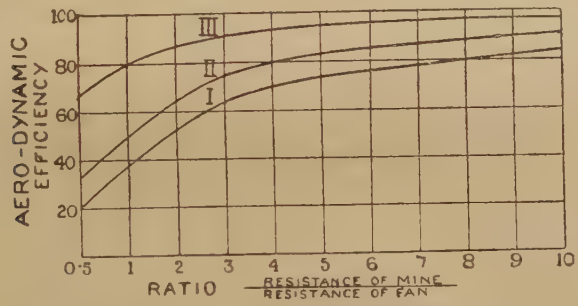


FIG. 191.—Efficiency of ventilating systems: (I) two identical fans in series; (II) single fan; (III) two identical fans in parallel.

When two fans run in parallel the quantity is increased over that given by one fan if the fan resistances are not negligible.

Paralleled Fans running at Dissimilar Speeds.—Pursuing the same line of reasoning as in the above commendable paper, and adhering to the same symbols, if it is assumed that dynamo A be kept at constant speed and B to be slowed down, the E.M.F. of B will fall, as also its current, while the current through A will rise above that obtaining when both had the same E.M.F. Ultimately no current will flow through B, and all through A, and if this current is I the voltage drop in A is I_1r_1, and the terminal voltage $E_1 - I_1r_1 = e_1$. As no current flows through B its E.M.F. must be equal and opposite to the terminal voltage of A, or $E_2 = e_1$.

The effect is the same as if B did not exist, and any further fall in the speed of B would cause reversal of the current in it and B would start "motoring."

With paralleled fans the analogy is very close to the above

case, the depression of the fan which is being slowed down falling off as the square of the speed reduction (see law *p*5), so that this fan takes less air and the other fan more air. Let the drift-gauge for only one fan be h and the total water gauge of each of the fans running at the same speed be h_a and h_b respectively, and let h_a be kept constant, while h_b falls as the speed of B is reduced. When both are running at the same speeds in parallel the drift gauge will be slightly greater than h, as seen in the electrical analogy given, and this will fall until it becomes h, when B is passing no air, so that by the same analogous reasoning h is the total depression produced by B. The depression produced by B will now be just sufficient to prevent A from pulling its air through B direct from the atmosphere, and any further fall in the speed of B would cause it to "motor," *i.e.* reverse. From the preceding discussion it will be apparent that *no greatly increased volume results from running two fans in parallel at the same speed as when operating singly, and only on the assumption that the internal resistance is high can it be shown that an increase does result.* Mr Joseph Parker in a few lucid papers before the Institution of Mining Engineers endeavours, very rightly, to destroy this constant-speed obsession prevalent among mining men, by advocating constant power or power regulation instead. This can be achieved on an electrical drive by a D.C. series-wound motor or a series characteristic single-phase motor, and on a steam drive by adjustable cut-off gear. He cites two first principles cases thus—

(*a*) The ventilating pressure varies as the square of the volume passing, so that doubling the volume means quadrupling the pressure. Now if two fans be paralleled on a constant mine resistance so as to pass through each fan the same volume as when working alone, the quantity will be doubled but the water gauge quadrupled. It will be seen that now each fan is producing the same quantity as before, but at four times the former water gauge; it is just the same as if the fan had to work on a mine having a resistance four times as great as that of the actual mine.

(*b*) Suppose a fan runs on a mine with an external water gauge of W.G. producing Q cub. ft. of air per minute, and that another fan passing half this quantity be added in parallel. The total volume passed will be $1\frac{1}{2}$Q, and the external water gauge will be increased to $(1\frac{1}{2})^2$ or $2\frac{1}{4}$ W.G. Now each fan is in the same position as if it had been put to work on a mine having a resistance $2\frac{1}{4}$ times as great as the actual mine resistance.

In case (*a*) the fans would develop equal powers, and the two fans would double the power of one of them, and in case (*b*) the combined powers would be one and a half times that developed by the most powerful fan. In each case the equivalent resistance on which the individual fans may be supposed to operate when they are paralleled will be the resistance of the mine increased in the proportion of the sum of the squares of the volumes which the fans could circulate separately. Hence "the equivalent resistance against which two fans running in parallel may be considered to operate is equal to 2^2 times the resistance of the mine R when the fans and motors are duplicates; and for other cases is n^2R, where n is the number of times the two fans are together more powerful than the fan running before being paralleled." * This law assumes that the fan-motor develops the same power after as before combination, otherwise the value of n varies.

The above writer gives, in the places cited, sufficient reasons to call for a reorganisation of the present system of calculating mine resistances, an attitude which we have shown previously to have long been adopted by Professor Briggs and other writers. Mr Parker suggests that, since the unit of *direct* resistance is immediately derivable from Atkinson's formula, it be adopted as the standard unit and called one *Atkinson*. A suitable measure of this unit would be that resistance which absorbs 10 lbs. of pressure per sq. ft. when a volume of 100,000 cub. ft. of air is passing per minute. An equivalent orifice of 10 sq. ft. might be called a *Murgue*. The above-discussed unit of *equivalent* resistance (n^2R) could be termed a *Garforth*.†

Problems on Combined Fans.

E.g. (i) Two identical fans, each capable of exhausting from a mine 200,000 cub. ft. of air per minute, are combined in series. What will be the quantity produced when—

(*a*) The water gauge is kept the same as when working singly?

(*b*) The power is the same as the sum of the single powers? (Assuming that the internal fan resistances and natural ventilation are negligible.)

(*a*) $200{,}000 \times \sqrt{2} = 282{,}000$ cub. ft. per minute.

(*b*) $200{,}000 \times \sqrt[3]{2} = 252{,}000$,, ,,

* Mr Joseph Parker, *Trans. Inst. M.E.*, lxiii. 224. † *Ibid.*, p. 234.

E.g. (ii) Two similar fans, each normally producing 200,000 cub. ft. per minute at a mine where the water gauge is $4\frac{1}{2}$ in., and each having internal resistances of $\frac{3}{4}$ in. W.G., are set to work in parallel on a mine and to maintain the same water gauge of $4\frac{1}{2}$ in. What is the resulting volume ?

The total equivalent resistance for one fan is =

$$R + r_1 = R + \frac{\frac{3}{4}}{4\frac{1}{2}}R = \frac{7}{6}R,$$

and the combined resistance of the two together is by (253)

$$\frac{r_1}{4} = \frac{1}{4} \times \frac{1}{6}R = \frac{1}{24}R, \text{ when they take the air equally}$$

$$\therefore \text{ the total equivalent resistance} = R + \frac{1}{24}R = \frac{25}{24}R,$$

and since h is constant ; then by (127) the combined quantity =

$$200{,}000 \sqrt{\frac{\frac{7}{6}}{\frac{25}{24}}} = 200000 \times 1{\cdot}05 = 210000, \textit{ i.e.}\ 105{,}000 \text{ cub. ft. per minute by each fan.}$$

E.g. (iii) Two similar fans, each capable of dealing with 300,000 cub. ft. per minute, and having internal resistance of 1 in. W.G. each, are run together in parallel on a mine where the external resistance is 4 in. W.G. One of the fans is slowed down so that it passes one-third of the quantity of its neighbour, which runs at constant speed. What will be the quantity passing through each fan ?

The fan resistances are each normally $\frac{1}{4}$th that of the mine, so the equivalent total is $= R + \frac{1}{4}R = \frac{5}{4}R$ for one alone.

Calling the smaller quantity x, the larger will be nx, and if the fan resistance coefficient be k, which resistance is assumed to vary as the square of the fan speed, the resistance of the one fan will be kx^2 and of the other $k(nx)^2$.

Combining these resistances, as in (252), and taking k as equal to unity, this becomes $\frac{n^2}{1+2n+n^2}$, which, since $n = 3$, becomes $\frac{9}{16}$. Thus the combined equivalent resistance is $\frac{9}{16}$ths of one fan alone. Therefore in order that one fan must pass three times as much air as the other the total combined resistance is $\frac{9}{16}$ times $\frac{1}{4}R$

or $\frac{9}{64}$ R. The total equivalent resistance is now $\frac{9}{64}R+R=\frac{73}{64}R$, and since quantity will vary as $\frac{1}{\sqrt{R}}$, as in the previous example,

$$Q_1=Q\sqrt{\frac{\frac{5}{4}}{\frac{73}{64}}}=Q\sqrt{\frac{80}{73}}=1{\cdot}047Q,$$ or 0·78525Q by the one and 0·26175Q by the other. This is 235.575 cub. ft. per minute and 78.525 cub. ft. per minute respectively.

E.g. (iv) In the foregoing example what is the total water gauge of the fan of slower speed?

Of the total water gauge $\frac{64}{73}$rds is spent on the mine and the remaining $\frac{9}{73}$rds on the fans and their connected approaches. Now the quantity passing through one fan is only one-third that passing through the other, consequently the water gauge absorbed in the former will be only one-ninth that absorbed in the other. The slowed-down fan has a total water gauge, therefore, of $(\frac{1}{9}\times\frac{9}{73})-\frac{64}{73}=\frac{65}{73}$rds of the water gauge of the fan running at constant speed.

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TABLE XIV.—*Hygrometric Tables.**

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
62	62	62·0	0·556	6·2	100	512·9
	61	60·1	0·520	5·8	94	513·1
	60	58·3	0·487	5·4	88	·3
	59	56·4	0·456	5·1	82	·5
	58	54·6	0·427	4·7	77	·7
	57	52·7	0·399	4·4	72	513·9
	56	50·8	0·272	4·1	67	514·1
	55	49·0	0·347	3·9	62	·2
	54	47·1	0·323	3·6	58	·4
	53	45·3	0·301	3·4	54	·5
	52	43·4	0·281	3·1	50	·7
	51	41·5	0·262	2·9	47	·8
	50	39·7	0·244	2·7	44	514·9
	49	37·8	0·227	2·5	41	515·0
	48	36·0	0·211	2·4	38	·1
	47	34·1	0·196	2·2	35	·2
	46	32·2	0·182	2·0	32	·3
	45	30·4	0·169	1·9	30	·4
	44	28·5	0·157	1·7	28	·5
	43	26·7	0·145	1·6	26	515·6
63	63	63·0	0·576	6·4	100	511·7
	62	61·2	0·540	6·0	94	512·0
	61	59·3	0·506	5·6	88	·2
	60	57·5	0·474	5·2	82	·4
	59	55·6	0·443	4·9	77	512·6
	58	53·8	0·414	4·6	72	513·0
	57	51·9	0·387	4·3	67	·1
	56	50·1	0·361	4·0	63	·1
	55	48·2	0·337	3·7	59	·3
	54	46·4	0·314	3·5	55	·4
	53	44·5	0·292	3·3	51	·6
	52	42·6	0·272	3·0	47	·7
	51	40·8	0·253	2·8	44	513·8
	50	39·0	0·236	2·6	41	514·0
	49	37·1	0·220	2·4	38	·1
	48	35·2	0·205	2·3	35	·2
	47	33·4	0·191	2·1	33	·3
	46	31·6	0·178	2·0	31	·3
	45	29·7	0·165	1·8	29	·4
	44	27·8	0·153	1·7	27	514·5
64	64	64·0	0·596	6·6	100	510·7
	63	62·2	0·560	6·2	94	510·9

* From the tables of the late Prof. James Glaisher, by the kind permission of Dr. J. W. L. Glaisher, F.R.S., Trinity College, Cambridge.

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn.=100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
64	62	60·3	0·525	5·8	88	511·1
	61	58·5	0·492	5·4	82	·4
	60	56·7	0·461	5·1	77	·6
	59	54·8	0·431	4·8	72	·8
	58	53·0	0·403	4·5	67	511·9
	57	51·2	0·376	4·2	63	512·1
	56	49·4	0·351	3·9	59	·3
	55	47·5	0·328	3·6	55	·4
	54	45·7	0·306	3·4	51	·6
	53	43·9	0·286	3·2	48	·7
	52	42·0	0·267	3·0	45	512·8
	51	40·2	0·249	2·8	42	513·0
	50	38·4	0·232	2·6	39	·1
	49	36·6	0·216	2·4	36	·2
	48	34·7	0·201	2·2	33	·3
	47	32·9	0·187	2·1	31	·4
	46	31·1	0·174	1·9	29	·5
	45	29·2	0·162	1·8	27	513·5
65	65	65·0	0·617	6·8	100	509·5
	64	63·2	0·580	6·4	94	509·8
	63	61·4	0·544	6·0	88	510·0
	62	59·5	0·510	5·6	83	·2
	61	57·7	0·478	5·3	78	·4
	60	55·9	0·447	4·9	73	·6
	59	54·1	0·418	4·6	68	510·8
	58	52·3	0·391	4·3	63	511·0
	57	50·4	0·366	4·0	59	·2
	56	48·6	0·342	3·8	55	·3
	55	46·8	0·320	3·5	51	·5
	54	45·0	0·299	3·3	48	·6
	53	43·2	0·279	3·1	45	·8
	52	41·3	0·260	2·9	42	511·9
	51	39·5	0·242	2·7	39	512·0
	50	37·7	0·225	2·5	36	·1
	49	35·9	0·210	2·3	34	·2
	48	34·1	0·196	2·2	32	·3
	47	32·2	0·182	2·0	29	·4
	46	30·4	0·169	1·9	27	·5
	45	28·6	0·157	1·7	25	·6
	44	26·8	0·146	1·6	23	512·6
66	66	66·0	0·639	7·0	100	508·5
	65	64·2	0·601	6·6	94	508·8
	64	62·4	0·564	6·2	88	509·0
	63	60·6	0·529	5·8	83	·2
	62	58·8	0·496	5·5	78	·4
	61	57·0	0·464	5·1	73	·6

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
66	60	55·1	0·434	4·8	68	509·8
	59	53·3	0·406	4·5	64	510·0
	58	51·5	0·380	4·2	60	·2
	57	49·7	0·356	3·9	56	·4
	56	47·9	0·333	3·7	52	·5
	55	46·1	0·311	3·4	48	·6
	54	44·3	0·280	3·2	45	·8
	53	42·5	0·271	3·0	42	510·9
	52	40·7	0·253	2·8	40	511·0
	51	38·9	0·236	2·6	37	·1
	50	37·0	0·220	2·4	34	·2
	49	35·2	0·205	2·3	32	·3
	48	33·4	0·191	2·1	30	·4
	47	31·6	0·178	2·0	28	·5
	46	29·8	0·165	1·8	26	·6
	45	28·0	0·153	1·7	24	511·7
67	67	67·0	0·661	7·3	100	507·4
	66	65·2	0·622	6·8	94	·6
	65	63·4	0·585	6·4	88	507·9
	64	61·6	0·549	6·0	83	508·1
	63	59·8	0·515	5·6	78	·3
	62	58·0	0·483	5·3	73	·5
	61	56·2	0·453	5·0	68	·7
	60	54 4	0·424	4·7	64	508·9
	59	52·6	0·397	4·4	60	509·1
	58	50·8	0·372	4·1	56	·2
	57	49·0	0·348	3·8	52	·4
	56	47·2	0·325	3·6	49	·5
	55	45·4	0·304	3·3	46	·7
	54	43·6	0·284	3·1	43	·8
	53	41·8	0·265	2·9	40	509·9
	52	40·0	0·247	2·7	37	510·1
	51	38·2	0·230	2·5	34	·2
	50	36·4	0·214	2·4	32	·3
	49	34·6	0·199	2·2	30	·4
	48	32·8	0·185	2·0	28	·4
	47	31·0	0·172	1·9	26	·5
	46	29·2	0·160	1·8	24	510·6
68	68	68·0	0·684	7·5	100	506·2
	67	66·2	0·644	7·1	94	·5
	66	64·4	0·606	6·6	88	506·7
	65	62·6	0·569	6·2	83	507·0
	64	60·8	0·534	5·8	78	·2
	63	59·1	0·501	5·5	73	·4
	62	57·3	0·470	5·2	68	·6
	61	55·5	0·441	4·8	64	507·8

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn.=100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
68	60	53·7	0·413	4·5	60	508·0
	59	51·9	0·387	4·2	56	·2
	58	50·1	0·362	4·0	52	·3
	57	48·3	0·339	3·7	49	·5
	56	46·5	0·317	3·5	46	·6
	55	44·7	0·296	3·2	43	·7
	54	42·9	0·276	3·0	40	508·9
	53	41·2	0·258	2·8	37	509·0
	52	39·4	0·241	2·6	35	·1
	51	37·6	0·225	2·5	33	·2
	50	35·8	0·210	2·3	31	·3
	49	34·0	0·196	2·2	29	·4
	48	32·2	0·182	2·0	27	·5
	47	30·4	0·169	1·9	25	·6
	46	28·6	0·157	1·7	23	509·6
69	69	69·0	0·707	7·8	100	505·2
	68	67·2	0·665	7·3	94	·4
	67	65·4	0·625	6·9	88	·7
	66	63·7	0·587	6·5	83	505·9
	65	61·9	0·551	6·1	78	506·2
	64	60.1	0·517	5·7	73	·4
	63	58·3	0·486	5·3	68	·6
	62	56·5	0·457	5·0	64	506·8
	61	54·8	0·429	4·7	60	507·0
	60	53·0	0·402	4·4	56	·1
	59	51·2	0·377	4·1	53	·3
	58	49·4	0·353	3·9	50	·5
	57	47·6	0·330	3·6	47	·6
	56	45·9	0·309	3·4	44	·7
	55	44·1	0·289	3·2	41	507·9
	54	42·3	0·270	3·0	38	508·0
	53	40·5	0·252	2·8	35	·1
	52	38·7	0·235	2·6	33	·2
	51	37·0	0·219	2·4	31	·3
	50	35·2	0·204	2·2	29	·4
	49	33·4	0·190	2·1	27	·5
	48	31·6	0·177	2·0	25	508·6
70	70	70·0	0·733	8·0	100	504·0
	69	68·2	0·691	7·5	94	·3
	68	66·5	0·651	7·1	88	·6
	67	64·7	0·613	6·7	83	504·8
	66	62·9	0·576	6·3	78	505·0
	65	61·1	0·541	5·9	73	·3
	64	59·4	0·508	5·5	69	·5
	63	57·6	0·476	5·2	65	·7
	62	55·8	0·446	4·9	61	505·9

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
70	61	54·0	0·418	4·6	57	506·0
	60	52·3	0·392	4·3	53	·2
	59	50·5	0·368	4·0	50	·4
	58	48·8	0·354	3·8	47	·5
	57	47·0	0·323	3·5	44	·7
	56	45·2	0·302	3·3	41	·8
	55	43·5	0·283	3·1	38	506·9
	54	41·7	0·265	2·9	36	507·1
	53	39·9	0·247	2·7	34	·2
	52	38·1	0·230	2·5	31	·3
	51	36·4	0·214	2·3	29	·4
	50	34·6	0·199	2·2	27	·5
	49	32·8	0·185	2·0	25	507·5
71	71	71·0	0·759	8·3	100	502·9
	70	69·2	0·714	7·8	94	503·2
	69	67·5	0·672	7·3	88	·4
	68	65·7	0·633	6·9	83	·7
	67	64·0	0·596	6·5	78	503·9
	66	62·2	0·560	6·1	73	504·2
	65	60·4	0·526	5·7	69	·4
	64	58·7	0·494	5·4	65	·6
	63	56·9	0·464	5·1	61	·7
	62	55·2	0·436	4·7	57	504·9
	61	53·4	0·409	4·4	53	505·1
	60	51·6	0·383	4·2	50	·3
	59	49·9	0·359	3·9	47	·5
	58	48·1	0·336	3·7	44	·6
	57	46·4	0·315	3·4	41	·7
	56	44·6	0·295	3·2	38	505·9
	55	42·8	0·276	3·0	36	506·0
	54	41·1	0·258	2·8	34	·1
	53	39·3	0·241	2·6	32	·2
	52	37·6	0·225	2·5	30	·3
	51	35·8	0·210	2·3	28	·4
	50	34·0	0·196	2·1	26	·5
	49	32·3	0·183	2·0	24	·6
	48	30·5	0·170	1·9	22	506·7
72	72	72·0	0·785	8·5	100	501·8
	71	70·2	0·739	8·0	94	502·1
	70	68·5	0·696	7·6	89	·4
	69	66·7	0·655	7·1	84	·6
	68	65·0	0·617	6·7	79	502·9
	67	63·2	0·581	6·3	74	503·1
	66	61·5	0·547	5·9	69	·4
	65	59·7	0·514	5·6	65	·6
	64	58·0	0·483	5·3	61	·8

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
72	63	56·2	0·454	5·0	57	503·9
	62	54·5	0·426	4·7	54	504·1
	61	52·7	0·399	4·4	51	·3
	60	51·0	0·374	4·1	48	·5
	59	49·2	0·350	3·8	45	·6
	58	47·5	0·328	3·6	42	·8
	57	45·7	0·307	3·3	39	504·9
	56	44·0	0·287	3·1	36	505·0
	55	42·2	0·269	2·9	34	·1
	54	40·5	0·252	2·7	32	·3
	53	38·7	0·236	2·6	30	·4
	52	37·0	0·221	2·4	28	·5
	51	35·3	0·207	2·2	26	·6
	50	33·5	0·193	2·0	24	·6
	49	31·8	0·180	1·9	23	505·7
73	73	73·0	0·812	8·8	100	500·7
	72	71·3	0·766	8·3	94	501·0
	71	69·5	0·722	7·8	89	·3
	70	67·8	0·680	7·4	84	·5
	69	66·0	0·641	7·0	79	501·8
	68	64·3	0·604	6·6	74	502·0
	67	62·6	0·568	6·2	70	·3
	66	60·8	0·534	5·8	66	·5
	65	59·1	0·502	5·4	62	·7
	64	57·3	0·472	5·1	58	502·9
	63	55·6	0·443	4·8	54	503·1
	62	53·9	0·416	4·5	51	·2
	61	52·1	0·390	4·2	48	·4
	60	50·4	0·366	4·0	45	·5
	59	48·6	0·343	3·7	42	·7
	58	46·9	0·322	3·5	40	503·8
	57	45·2	0·302	3·3	37	504·0
	56	43·4	0·283	3·1	35	·1
	55	41·7	0·265	2·9	32	·2
	54	39·9	0·248	2·7	30	·3
	53	38·2	0·232	2·5	28	·4
	52	36·5	0·217	2·3	26	·5
	51	34·7	0·202	2·2	24	·6
	50	33·0	0·188	2·0	23	504·7
74	74	74·0	0·840	9·1	100	499·6
	73	72·3	0·793	8·6	94	·9
	72	70·5	0·748	8·1	89	500·2
	71	68·8	0·705	7·6	84	·5
	70	67·1	0·664	7·2	79	500·7
	69	65·3	0·625	6·8	74	501·0
	68	63·6	0·588	6·4	70	·2

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
74	67	61·9	0·554	6·0	66	501·5
	66	60·2	0·522	5·6	62	·7
	65	58·4	0·492	5·3	58	501·9
	64	56·7	0·463	5·0	55	502·1
	63	55·0	0·435	4·7	52	·2
	62	53·2	0·408	4·4	48	502·4
	61	51·5	0·382	4·1	45	·6
	60	49·8	0·358	3·9	43	·7
	59	48·1	0·336	3·6	40	502·9
	58	46·3	0·315	3·4	37	503·0
	57	44·6	0·295	3·2	35	·1
	56	42·9	0·276	3·0	33	·2
	55	41·1	0·258	2·8	31	·3
	54	39·4	0·241	2·6	29	·4
	53	37·7	0·225	2·4	27	·5
	52	35·9	0·210	2·3	25	·6
	51	34·2	0·196	2·1	23	503·7
75	75	75·0	0·868	9·4	100	498·5
	74	73·3	0·820	8·9	94	498·8
	73	71·6	0·774	8·4	89	499·1
	72	69·8	0·731	7·9	84	·4
	71	68·1	0·690	7·4	79	·6
	70	66·4	0·650	7·0	74	499·9
	69	64·7	0·612	6·6	70	500·1
	68	63·0	0·576	6·2	66	·4
	67	61·2	0·542	5·8	62	·6
	66	59·5	0·510	5·5	58	500·8
	65	57·8	0·479	5·2	55	501·0
	64	56·1	0·450	4·9	52	·2
	63	54·4	0·423	4·6	49	·3
	62	52·6	0·397	4·3	46	·5
	61	50·9	0·373	4·0	43	·6
	60	49·2	0·350	3·8	40	·8
	59	47·5	0·328	3·6	38	501·9
	58	45·8	0·307	3·3	36	502·1
	57	44·0	0·288	3·1	33	·2
	56	42·3	0·270	2·9	31	·3
	55	40·6	0·253	2·7	29	·4
	54	38·9	0·237	2·6	27	·5
	53	37·2	0·222	2·4	25	502·6
76	76	76·0	0·987	9·7	100	497·4
	75	74·3	0·848	9·2	94	497·7
	74	72·6	0·801	8·6	89	498·0
	73	70·9	0·756	8·2	84	·3
	72	69·2	0·713	7·7	79	·5
	71	67·4	0·672	7·2	75	498·8

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
76	70	65·7	0·633	6·8	71	499·0
	69	64·0	0·596	6·4	67	·3
	68	62·3	0·561	6·1	63	·5
	67	60·6	0·528	5·7	59	·4
	66	58·9	0·497	5·4	55	499·9
	65	57·2	0·468	5·1	52	500·1
	64	55·5	0·441	4·8	49	·3
	63	53·8	0·415	4·5	46	·4
	62	52·1	0·390	4·2	43	·6
	61	50·4	0·366	3·9	40	·7
	60	48·6	0·343	3·7	38	500·9
	59	46·9	0·322	3·5	36	501·0
	58	45·2	0·302	3·3	34	·2
	57	43·5	0·283	3·1	32	·3
	56	41·8	0·265	2·9	30	·4
	55	40·1	0·248	2·7	28	·5
	54	38·4	0·232	2·5	26	·6
	53	36·7	0·217	2·3	24	501·7
77	77	77·0	0·927	10·0	100	496·3
	76	75·3	0·877	9·5	94	·6
	75	73·6	0·829	8·9	89	496·9
	74	71·9	0·783	8·4	84	497·2
	73	70·2	0·739	8·0	79	·5
	72	68·5	0·697	7·5	75	497·8
	71	66·8	0·658	7·1	71	498·0
	70	65·1	0·620	6·7	67	·3
	69	63·4	0·584	6·3	63	·5
	68	61·7	0·550	5·9	59	·7
	67	60·0	0·518	5·6	56	498·9
	66	58·3	0·488	5·3	53	499·1
	65	56·6	0·459	4·9	50	·3
	64	54·9	0·431	4·6	47	·4
	63	53·2	0·405	4·3	44	·6
	62	51·5	0·381	4·1	41	·8
	61	49·8	0·358	3·9	38	499·9
	60	48·1	0·336	3·6	36	500·1
	59	46·4	0·315	3·4	34	·2
	58	44·7	0·295	3·2	32	·3
	57	43·0	0·276	3·0	30	·4
	56	41·3	0·259	2·8	28	·5
	55	39·6	0·243	2·6	26	·6
	54	37·9	0·228	2·4	24	500·7
78	78	78·0	0·958	10·3	100	495·2
	77	76·3	0·906	9·7	94	·5
	76	74·6	0·857	9·2	89	495·8
	75	72·9	0·810	8·7	84	496·1

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn.=100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
°F.	°F.	° F.	In. Hg.	Grains.		Grains.
78	74	71·2	0·765	8·2	79	496·4
	73	69·5	0·722	7·8	75	·7
	72	67·8	0·681	7·3	71	496·9
	71	66·1	0·642	6·9	67	497·2
	70	64·4	0·605	6·5	63	·4
	69	62·7	0·571	6·2	59	·6
	68	61·1	0·539	5·8	56	497·8
	67	59·4	0·508	5·5	53	498·0
	66	57·7	0·478	5·1	50	·2
	65	56·0	0·449	4·8	47	·4
	64	54·3	0·422	4·5	44	·5
	63	52·6	0·397	4·3	41	·7
	62	50·9	0·373	4·0	39	498·8
	61	49·2	0·350	3·8	37	499·0
	60	47·5	0·328	3·5	35	·1
	59	45·8	0·308	3·3	32	·3
	58	44·1	0·289	3·1	30	·4
	57	42·4	0·271	2·9	28	·5
	56	40·7	0·254	2·7	27	·6
	55	39·0	0·238	2·5	25	499·7
79	79	79·0	0·999	10·6	100	494·1
	78	77·3	0·937	10·1	95	·5
	77	75·6	0·887	9·5	90	494·8
	76	73·9	0·839	9·0	85	495·1
	75	72·3	0·793	8·5	80	·4
	74	70·6	0·749	8·0	75	·7
	73	68·9	0·707	7·6	71	495·9
	72	67·2	0·666	7·2	67	496·2
	71	65·5	0·628	6·8	63	·4
	70	63·8	0·592	6·4	59	·6
	69	62·1	0·558	6·0	56	496·8
	68	60·4	0·526	5·6	53	497·0
	67	58·7	0·495	5·3	50	·2
	66	57·0	0·466	5·0	47	·4
	65	55·4	0·439	4·7	44	·6
	64	53·7	0·413	4·4	42	·8
	63	52·0	0·388	4·2	39	497·9
	62	50·3	0·365	3·9	37	498·1
	61	48·6	0·343	3·7	35	·2
	60	46·9	0·322	3·5	32	·3
	59	45·2	0·302	3·2	30	·5
	58	43·5	0·283	3·0	28	·6
	57	41·8	0·265	2·8	26	·7
	56	40·2	0·248	2·7	25	498·8
80	80	80·0	1·023	11·0	100	493·0
	79	78·3	0·968	10·4	95	·4

TABLE XIV.—*Hygrometric Tables* continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
80	78	76·6	0·916	9·8	90	493·7
	77	75·0	0·867	9·3	85	494·0
	76	73·3	0·820	8·8	80	·3
	75	71·6	0·775	8·3	75	·6
	74	69·9	0·732	7·8	71	494·8
	73	68·2	0·690	7·4	67	495·1
	72	66·5	0·650	7·0	63	·3
	71	64·9	0·613	6·6	59	·6
	70	63·2	0·578	6·2	56	495·8
	69	61·5	0·545	5·8	53	496·0
	68	59·8	0·513	5·5	50	·2
	67	58·1	0·483	5·2	47	·4
	66	56·4	0·455	4·9	44	·6
	65	54·8	0·429	4·6	41	·7
	64	53·1	0·404	4·3	39	496·9
	63	51·4	0·380	4·1	37	497·0
	62	49·7	0·357	3·8	35	·2
	61	48·0	0·335	3·6	33	·3
	60	46·3	0·315	3·4	31	·4
	59	44·7	0·296	3·2	29	·6
	58	43·0	0·278	3·0	27	·7
	57	41·3	0·261	2·8	25	497·8
81	81	81·0	1·057	11·3	100	491·8
	80	79·3	1·000	10·7	95	492·2
	79	77·7	0·947	10·1	90	·5
	78	76·0	0·897	9·5	85	492·8
	77	74·3	0·849	9·1	80	493·1
	76	72·6	0·802	8·6	76	·4
	75	70·9	0·757	8·1	72	493·7
	74	69·3	0·715	7·6	68	494·0
	73	67·6	0·675	7·2	64	·2
	72	65·9	0·637	6·8	60	·5
	71	64·2	0·601	6·4	56	·7
	70	62·6	0·567	6·0	53	494·9
	69	60·9	0·534	5·7	50	495·1
	68	59·2	0·503	5·4	47	·3
	67	57·5	0·473	5·1	44	·5
	66	55·8	0·445	4·8	41	·6
	65	54·2	0·419	4·5	39	495·8
	64	52·5	0·394	4·2	37	496·0
	63	50·8	0·371	4·0	35	·1
	62	49·1	0·349	3·7	33	·3
	61	47·5	0·328	3·5	31	·4
	60	45·8	0·308	3·3	29	·5
	59	44·1	0·289	3·1	27	·6
	58	42·4	0·271	2·9	26	496·7

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
82	82	82·0	1·091	11·7	100	490·8
	81	80·3	1·033	11·1	95	491·1
	80	78·7	0·978	10·5	90	·5
	79	77·0	0·926	9·9	85	491·8
	78	75·3	0·877	9·4	80	492·1
	77	73·6	0·830	8·9	76	·4
	76	72·0	0·785	8·4	72	492·7
	75	70·3	0·742	7·9	68	493·0
	74	68·6	0·701	7·5	64	·2
	73	67·0	0·662	7·1	60	·5
	72	65·3	0·624	6·7	57	·7
	71	63·6	0·588	6·3	54	493·9
	70	61·9	0·554	5·9	51	494·1
	69	60·3	0·522	5·6	48	·3
	68	58·6	0·492	5·2	45	·5
	67	56·9	0·464	4·9	42	·7
	66	55·3	0·437	4·7	40	494·8
	65	53·6	0·411	4·4	38	495·0
	64	51·9	0·387	4·1	35	·2
	63	50·2	0·364	3·9	33	·3
	62	48·6	0·342	3·6	31	·5
	61	46·9	0·321	3·4	29	·6
	60	45·2	0·301	3·2	27	·7
	59	43·5	0·282	3·0	26	495·8
83	83	83·0	1·127	12·0	100	489·6
	82	81·3	1·067	11·7	95	490·0
	81	79·7	1·010	10·8	90	·4
	80	78·0	0·956	10·2	85	490·7
	79	76·3	0·905	9·7	80	491·0
	78	74·7	0·856	9·1	76	·3
	77	73·0	0·810	8·6	72	·6
	76	71·3	0·766	8·2	68	491·8
	75	69·7	0·724	7·7	64	492·1
	74	68·0	0·684	7·3	60	·4
	73	66·3	0·646	6·9	57	·6
	72	64·7	0·610	6·5	54	492·9
	71	63·0	0·576	6·1	51	493·1
	70	61·3	0·543	5·8	48	·3
	69	59·7	0·512	5·4	45	·5
	68	58·0	0·482	5·1	42	·6
	67	56·3	0·454	4·8	40	493·8
	66	54·7	0·428	4·6	38	494·0
	65	53·0	0·403	4·3	36	·1
	64	51·3	0·379	4·0	34	·3
	63	49·7	0·356	3·8	32	·4
	62	48·0	0·335	3·6	30	·6

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn.=100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
83	61	46·3	0·315	3·3	28	494·7
	60	44·7	0·296	3·1	26	494·8
84	84	84·0	1·164	12·4	100	488·4
	83	82·3	1·103	11·7	95	488·8
	82	80·7	1·045	11·1	90	489·2
	81	79·0	0·990	10·5	85	·6
	80	77·4	0·938	10·0	80	489·9
	79	75·7	0·888	9·4	76	490·2
	78	74·0	0·840	8·9	72	·5
	77	72·4	0·794	8·5	68	490·8
	76	70·7	0·751	8·0	64	491·1
	75	69·1	0·710	7·5	60	·3
	74	67·4	0·671	7·1	57	·5
	73	65·7	0·634	6·7	54	491·8
	72	64·1	0·598	6·3	51	492·0
	71	62·4	0·564	6·0	48	·2
	70	60·8	0·532	5·6	45	·4
	69	59·1	0·501	5·3	43	·6
	68	57·4	0·472	5·0	41	·8
	67	55·8	0·445	4·7	38	492·9
	66	54·1	0·419	4·5	36	493·1
	65	52·5	0·394	4·2	34	·2
	64	50·8	0·371	4·0	32	·4
	63	49·1	0·349	3·7	30	·5
	62	47·5	0·328	3·5	28	·7
	61	45·8	0·308	3·3	26	493·8
85	85	85·0	1·203	12·8	100	487·4
	84	83·4	1·141	12·1	95	487·8
	83	81·7	1·082	11·5	90	488·2
	82	80·0	1·026	10·9	85	·5
	81	78·4	0·973	10·3	80	488·8
	80	76·7	0·922	9·7	76	489·2
	79	75·1	0·873	9·2	72	·5
	78	73·4	0·826	8·7	68	489·8
	77	71·8	0·781	8·3	64	490·0
	76	70·1	0·738	7·8	61	·3
	75	68·5	0·698	7·4	58	·6
	74	66·8	0·660	7·0	55	490·8
	73	65·2	0·623	6·6	52	491·0
	72	63·5	0·588	6·2	49	·2
	71	61·8	0·555	5·9	46	·4
	70	60·2	0·524	5·5	43	·6
	69	58·5	0·494	5·2	40	491·8
	68	56·9	0·466	4·9	38	492·0
	67	55·2	0·439	4·6	36	·2
	66	53·6	0·413	4·3	34	3

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn.=100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
85	65	51·9	0·388	4·1	32	492·5
	64	50·2	0·365	3·8	30	·6
	63	48·6	0·343	3·6	28	·8
	62	46·9	0·322	3·4	27	492·9
86	86	86·0	1·242	13·2	100	486·2
	85	84·3	1·180	12·5	95	486·6
	84	82·7	1·121	11·8	90	487·0
	83	81·1	1·064	11·2	85	·4
	82	79·4	1·008	10·6	80	487·7
	81	77·8	0·955	10·1	76	488·1
	80	76·1	0·904	9·5	72	·4
	79	74·5	0·855	9·0	68	488·7
	78	72·8	0·808	8·5	64	489·0
	77	71·2	0·763	8·1	61	·2
	76	69·5	0·721	7·6	58	489·5
	75	67·9	0·681	7·2	55	·7
	74	66·2	0·643	6·8	52	490·0
	73	64·6	0·607	6·4	49	·2
	72	62·9	0·573	6·1	46	·4
	71	61·3	0·541	5·7	43	·6
	70	59·6	0·510	5·4	40	490·8
	79	58·0	0·480	5·1	38	491·0
	68	56·3	0·452	4·8	36	·1
	67	54·7	0·426	4·5	34	·3
	66	53·0	0·402	4·2	32	·4
	65	51·4	0·379	4·0	30	·6
	64	49·7	0·357	3·7	28	·7
	63	48·1	0·336	3·5	27	491·8
87	87	87·0	1·282	13·6	100	485·2
	86	85·4	1·219	12·9	95	485·6
	85	83·7	1·158	12·2	90	486·0
	84	82·1	1·098	11·6	85	·4
	83	80·4	1·040	11·0	81	486·7
	82	78·8	0·985	10·4	77	487·1
	81	77·1	0·933	9·8	73	·4
	80	75·5	0·884	9·3	69	487·7
	79	73·9	0·837	8·8	65	488·0
	78	72·2	0·792	8·3	61	·2
	77	70·6	0·749	7·9	58	·5
	76	68·9	0·708	7·4	55	488·8
	75	67·3	0·669	7·0	52	489·0
	74	65·6	0·632	6·6	49	·2
	73	64·0	0·597	6·3	46	·4
	72	62·4	0·564	5·9	43	·6
	71	60·7	0·532	5·6	41	489·8
	70	59·1	0·502	5·3	39	490·0

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
87	69	57·4	0·473	5·0	37	490·2
	68	55·8	0·446	4·7	35	·3
	67	54·1	0·420	4·4	33	·5
	66	52·5	0·395	4·2	31	·7
	65	50·8	0·372	3·9	29	·8
	64	49·2	0·351	3·7	27	490·9
88	88	88·0	1·323	14·0	100	484·0
	87	86·4	1·258	13·3	95	·4
	86	84·7	1·195	12·6	90	484·8
	85	83·1	1·134	12·0	85	485·2
	84	81·5	1·075	11·4	81	·6
	83	79·8	1·018	10·8	77	485·9
	82	78·2	0·964	10·2	73	486·3
	81	76·5	0·913	9·6	69	·6
	80	74·9	0·865	9·1	65	486·9
	79	73·3	0·819	8·6	61	487·2
	78	71·6	0·775	8·1	58	·4
	77	70·0	0·733	7·7	55	·7
	76	68·4	0·693	7·3	52	487·9
	75	66·7	0·655	6·9	49	488·2
	74	65·1	0·619	6·6	46	·4
	73	63·4	0·585	6·1	43	·6
	72	61·8	0·553	5·8	41	488·8
	71	60·2	0·522	5·5	39	489·0
	70	58·5	0·492	5·2	37	·2
	69	56·9	0·464	4·9	35	·3
	68	55·2	0·437	4·6	33	·5
	67	53·6	0·411	4·3	31	·6
	66	52·0	0·387	4·1	29	·8
	65	50·3	0·365	3·8	27	489·9
89	89	89·0	1·366	14·4	100	482·9
	88	87·4	1·299	13·7	95	483·3
	87	85·7	1·234	13·0	90	483·7
	86	84·1	1·172	12·3	85	484·1
	85	82·5	1·112	11·7	81	·5
	84	80·8	1·054	11·1	77	484·8
	83	79·2	0·998	10·5	73	485·1
	82	77·6	0·945	10·0	69	·5
	81	76·0	0·895	9·4	65	485·8
	80	74·3	0·847	8·9	61	486·1
	79	72·7	0·801	8·4	58	·3
	78	71·1	0·757	8·0	55	·6
	77	69·4	0·716	7·5	52	486·9
	76	67·8	0·677	7·1	49	487·1
	75	66·2	0·640	6·7	46	·3
	74	64·6	0·605	6·4	43	·5

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn.=100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
89	73	63·0	0·572	6·0	41	487·7
	72	61·3	0·541	5·7	39	487·9
	71	59·6	0·511	5·4	37	488·1
	70	58·0	0·482	5·1	35	·3
	69	56·4	0·455	4·8	33	·4
	68	54·7	0·429	4·5	31	·6
	67	53·1	0·404	4·2	29	·8
	66	51·5	0·381	4·0	28	488·9
90	90	90·0	1·411	14·8	100	481·8
	89	88·4	1·342	14·1	95	482·3
	88	86·8	1·276	13·4	90	482·7
	87	85·1	1·212	12·7	85	483·1
	86	83·5	1·151	12·1	81	·4
	85	81·9	1·092	11·4	77	483·8
	84	80·3	1·036	10·8	73	484·1
	83	78·6	0·982	10·3	69	·5
	82	77·0	0·930	9·7	65	484·8
	81	75·4	0·880	9·2	62	485·1
	80	73·7	0·833	8·7	59	·4
	79	72·1	0·788	8·3	56	·6
	78	70·5	0·745	7·8	53	485·9
	77	68·8	0·704	7·4	50	486·1
	76	67·2	0·665	7·0	47	·4
	75	65·6	0·629	6·6	44	·6
	74	64·0	0·595	6·2	42	486·8
	73	62·4	0·562	5·9	40	487·0
	72	60·7	0·531	5·6	38	·2
	71	59·1	0·501	5·3	36	·4
	70	57·5	0·473	5·0	34	·5
	69	55·8	0·446	4·7	32	·7
	68	54·2	0·421	4·4	30	487·8
	67	52·6	0·397	4·2	28	488·0
91	91	91·0	1·455	15·3	100	480·7
	90	89·4	1·384	14·5	95	481·1
	89	87·8	1·316	13·8	90	·5
	88	86·1	1·250	13·1	86	481·9
	87	84·5	1·187	12·3	82	482·3
	86	82·9	1·127	11·8	78	482·6
	85	81·3	1·069	11·2	74	483·0
	84	79·7	1·013	10·6	70	·4
	83	78·0	0·960	10·1	66	483·7
	82	76·4	0·910	9·5	62	484·0
	81	74·8	0·862	9·0	59	·3
	80	73·2	0·816	8·5	56	·5
	79	71·6	0·772	8·1	53	484·8
	78	69·9	0·730	7·7	50	485·1

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
91	77	68·3	0·690	7·2	47	485·3
	76	66·7	0·652	6·8	44	·5
	75	65·1	0·616	6·5	42	485·8
	74	63·5	0·582	6·1	40	486·0
	73	61·8	0·550	5·8	38	·1
	72	60·2	0·520	5·5	36	·3
	71	58·6	0·491	5·1	34	·5
	70	57·0	0·463	4·8	32	·7
	69	55·3	0·437	4·5	30	486·8
	68	53·7	0·413	4·3	28	487·0
92	92	92·0	1·501	15·7	100	479·5
	91	90·4	1·428	14·9	95	480·0
	90	88·8	1·357	14·2	90	·4
	89	87·1	1·289	13·5	85	480·8
	88	85·5	1·224	12·8	81	481·2
	87	83·9	1·162	12·2	77	·5
	86	82·3	1·103	11·6	73	481·9
	85	80·7	1·047	11·0	70	482·3
	84	79·1	0·994	10·4	66	482·6
	83	77·5	0·943	9·9	62	·9
	82	75·9	0·894	9·3	59	483·2
	81	74·2	0·847	8·8	56	·5
	80	72·6	0·802	8·3	53	483·7
	79	71·0	0·759	7·9	50	484·0
	78	69·4	0·718	7·5	47	·2
	77	67·8	0·679	7·1	45	·5
	76	66·2	0·642	6·7	43	·7
	75	64·5	0·607	6·3	41	484·9
	74	62·9	0·573	6·0	38	485·1
	73	61·3	0·541	5·7	36	·3
	72	59·7	0·511	5·3	34	·5
	71	58·1	0·483	5·0	32	·6
	70	56·5	0·456	4·8	30	485·8
	69	54·8	0·430	4·5	28	486·0
93	93	93·0	1·548	16·2	100	478·4
	92	91·4	1·473	15·4	95	478·9
	91	89·8	1·401	14·7	90	479·3
	90	88·2	1·332	14·0	86	479·7
	89	86·6	1·266	13·3	82	480·1
	88	85·0	1·203	12·6	78	·5
	87	83·4	1·142	11·9	74	480·9
	86	81·7	1·084	11·3	70	481·2
	85	80·1	1·028	10·7	66	·6
	84	78·5	0·974	10·2	63	481·9
	83	76·9	0·923	9·6	60	482·2
	82	75·3	0·874	9·1	57	·5

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
93	81	73·7	0·828	8·7	54	482·8
	80	72·1	0·785	8·2	51	483·0
	79	70·5	0·744	7·8	48	·3
	78	68·9	0·704	7·4	45	·5
	77	67·2	0·666	6·9	42	483·7
	76	65·6	0·630	6·6	40	484·0
	75	64·0	0·596	6·2	38	·2
	74	62·4	0·564	5·9	36	·4
	73	60·8	0·533	5·5	34	·5
	72	59·2	0·503	5·2	32	·7
	71	57·6	0·475	5·0	30	484·9
	70	56·0	0·449	4·7	29	485·4
94	94	94·0	1·597	16·7	100	477·2
	93	92·4	1·520	15·9	95	477·7
	92	90·8	1·446	15·1	90	478·1
	91	89·2	1·375	14·4	86	478·6
	90	87·6	1·307	13·7	82	479·0
	89	86·0	1·242	13·0	78	·4
	88	84·4	1·180	12·3	74	479·7
	87	82·8	1·121	11·7	70	480·1
	86	81·2	1·064	11·1	66	·5
	85	79·6	1·010	10·5	63	480·8
	84	78·0	0·958	10·0	60	481·1
	83	76·4	0·909	9·5	57	·4
	82	74·8	0·862	9·0	54	481·7
	81	73·1	0·816	8·5	51	482·0
	80	71·5	0·772	8·0	48	·2
	79	69·9	0·731	7·6	45	·5
	78	68·3	0·692	7·2	43	·7
	77	66·7	0·655	6·8	41	482·9
	76	65·1	0·620	6·4	39	483·1
	75	63·5	0·587	6·1	37	·3
	74	61·9	0·555	5·7	35	·5
	73	60·3	0·524	5·4	33	·7
	72	58·7	0·494	5·1	31	483·9
	71	57·1	0·466	4·9	29	484·0
95	95	95·0	1·646	17·2	100	476·2
	94		1·567	16·3	95	476·6
	93	91·8	1·492	15·5	91	477·1
	92	90·2	1·420	14·8	86	·5
	91	88·6	1·350	14·1	82	477·9
	90	87·0	1·283	13·4	78	478·3
	89	85·4	1·219	12·7	74	478·7
	88	83·8	1·158	12·1	70	479·1
	87	82·2	1·099	11·5	66	·5
	86	80·6	1·043	10·9	63	479·8

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
95	85	79·0	0·990	10·3	60	480·1
	84	77·4	0·940	9·8	57	·4
	83	75·8	0·892	9·3	54	480·7
	82	74·2	0·846	8·8	51	481·0
	81	72·6	0·802	8·3	48	·3
	80	71·0	0·759	7·9	45	·5
	79	69·4	0·718	7·5	43	481·7
	78	67·8	0·680	7·1	41	482·0
	77	66·2	0·644	6·7	39	·2
	76	64·6	0·610	6·3	37	·4
	75	63·0	0·577	6·0	35	·6
	74	61·4	0·545	5·6	33	482·8
	73	59·8	0·514	5·3	31	483·0
	72	58·2	0·485	5·0	30	483·1
96	96	96·0	1·697	17·7	100	475·0
	95	94·4	1·616	16·8	95	·5
	94	92·8	1·538	16·0	90	475·9
	93	91·2	1·464	15·2	86	476·4
	92	89·6	1·393	14·5	82	476·8
	91	88·0	1·325	13·8	78	477·2
	90	86·4	1·260	13·1	74	477·6
	89	84·9	1·198	12·4	70	478·0
	88	83·3	1·138	11·8	66	·4
	87	81·7	1·081	11·2	63	478·7
	86	80·1	1·026	10·7	60	479·0
	85	78·5	0·974	10·1	57	·3
	84	76·9	0·924	9·6	54	·6
	83	75·3	0·876	9·1	52	479·9
	82	73·7	0·831	8·6	49	480·2
	81	72·1	0·788	8·2	46	·5
	80	70·5	0·746	7·7	43	·7
	79	68·9	0·706	7·3	41	480·9
	78	67·3	0·668	6·9	39	481·2
	77	65·7	0·633	6·5	37	·4
	76	64·2	0·600	6·2	35	·6
	75	62·6	0·568	5·9	33	·8
	74	61·0	0·537	5·6	31	481·9
	73	59·4	0·507	5·3	[illegible]	482·1
97	97	97·0	1·750	18·2	100	473·8
	96	95·4	1·666	17·3	95	474·3
	95	93·8	1·586	16·5	90	474·7
	94	92·2	1·510	15·7	86	475·2
	93	90·6	1·437	14·9	82	475·6
	92	89·1	1·368	14·2	78	476·1
	91	87·1	1·302	13·5	74	·5
	90	85·9	1·238	12·8	70	476·8

TABLE XIV.—*Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
97	89	84·3	1·177	12·2	67	477·2
	88	82·7	1·118	11·6	64	·6
	87	81·1	1·062	11·0	60	477·9
	86	79·5	1·009	10·4	57	478·2
	85	78·0	0·958	9·9	54	·5
	84	76·4	0·909	9·4	52	478·8
	83	74·8	0·863	8·9	49	479·1
	82	73·2	0·818	8·4	46	·4
	81	71·6	0·775	8·0	44	·7
	80	70·0	0·734	7·6	42	479·9
	79	68·4	0·695	7·2	39	480·1
	78	66·8	0·658	6·8	37	·3
	77	65·3	0·623	6·4	35	·5
	76	63·7	0·590	6·1	33	·7
	75	62·1	0·558	5·8	31	480·9
	74	60·5	0·528	5·5	30	481·1
98	98	98·0	1·807	18·7	100	472·7
	97	96·4	1·721	17·8	95	473·2
	96	94·8	1·639	17·0	90	473·7
	95	93·3	1·561	16·2	86	474·2
	94	91·7	1·487	15·4	82	474·6
	93	90·1	1·416	14·6	78	475·0
	92	88·5	1·347	13·9	74	·4
	91	86·9	1·281	13·2	70	475·8
	90	85·3	1·218	12·6	67	476·2
	89	83·8	1·157	12·0	64	·6
	88	82·2	1·099	11·4	61	476·9
	87	80·6	1·043	10·8	58	477·3
	86	79·0	0·990	10·2	55	·6
	85	77·4	0·940	9·7	52	477·9
	84	75·9	0·893	9·2	49	478·2
	83	74·3	0·848	8·7	46	·4
	82	72·7	0·804	8·3	44	·7
	81	71·1	0·762	7·8	42	478·9
	80	69·5	0·722	7·4	40	479·2
	79	68·0	0·684	7·1	38	·4
	78	66·4	0·648	6·7	36	·6
	77	64·8	0·613	6·3	34	479·8
	76	63·2	0·580	6·0	32	480·0
	75	61·6	0·548	5·7	30	480·2
99	99	99·0	1·862	19·3	100	471·5
	98	97·4	1·774	18·4	95	472·0
	97	95·9	1·690	17·5	91	472·5
	96	94·3	1·610	16·7	87	473·0
	95	92·7	1·534	15·9	83	·5
	94	91·1	1·462	15·1	79	473·9

TABLE XIV. — *Hygrometric Tables*—continued.

Reading of thermometer.		Temperature of the dew-point.	Elastic force of vapour.	Vapour in a cubic foot of air.	Degree of humidity. Satn. = 100.	Weight of a cub. ft. of air. Bar. reading 29 inches.
Dry.	Wet.					
° F.	° F.	° F.	In. Hg.	Grains.		Grains.
99	93	89·6	1·393	14·4	75	474·3
	92	88·0	1·327	13·7	71	474·7
	91	86·4	1·263	13·0	67	475·1
	90	84·4	1·201	12·3	64	·5
	89	83·2	1·141	11·7	61	475·9
	88	81·7	1·084	11·1	58	476·2
	87	80·1	1·029	10·5	55	·5
	86	78·5	0·977	10·0	52	476·8
	85	76·9	0·927	9·5	49	477·1
	84	75·4	0·880	9·0	46	·4
	83	73·8	0·835	8·6	44	·6
	82	72·2	0·792	8·1	42	477·9
	81	70·6	0·751	7·7	40	478·1
	80	69·1	0·712	7·3	38	·4
	79	67·5	0·675	6·9	36	·6
	78	65·9	0·639	6·6	34	478·8
	77	64·3	0·605	6·2	32	479·0
	76	62·8	0·572	5·9	31	479·2
100	100	100·0	1·918	19·8	100	470·5
	99	98·4	1·828	18·9	95	471·0
	98	96·9	1·742	18·0	90	471·5
	97	95·3	1·660	17·2	86	472·0
	96	93·7	1·582	16·3	82	·4
	95	92·1	1·508	15·5	78	472·9
	94	90·6	1·437	14·8	74	473·3
	93	89·0	1·368	14·1	71	473·7
	92	87·4	1·301	13·4	68	474·1
	91	85·9	1·237	12·7	64	·5
	90	84·3	1·175	12·1	61	474·9
	89	82·7	1·116	11·5	58	475·2
	88	81·2	1·060	10·9	55	·5
	87	79·6	1·006	10·4	52	475·8
	86	78·0	0·955	9·9	49	476·1
	85	76·5	0·907	9·4	47	·4
	84	74·9	0·861	8·9	45	·7
	83	73·3	0·818	8·4	43	476·9
	82	71·7	0·777	8·0	40	477·2
	81	70·2	0·738	7·6	38	·4
	80	68·6	0·700	7·2	36	·6
	79	67·0	0·663	6·8	34	477·9
	78	65·5	0·628	6·4	32	478·1
	77	63·9	0·594	6·1	30	·3
	76	62·4	0·561	5·8	29	478·5

INDEX.

Absolute pressure, 25.
— temperature, 25.
Absorption, 88.
Acceleration, defined, 4.
— of air through the fan, 292.
Acids, 62.
Adiabatic conditions, 7.
Adsorption, 86, 89.
Air constant (R), 43.
— filter, 185.
— locks, 256.
— measurement, 37.
— -methane mixtures, 77.
— — — temperature and pressure effects upon, 78.
— movement, 117.
— of mines, 86.
— system efficiency, 180.
Alkalies, 62.
Anemometer, 35.
Aneroid, normal and recording, 26.
Ankylostomiasis, 105.
Aqueous vapour, psychrometric formula, 116.
— — (see also water-vapour), 65.
Atkinson formula, 209.
— origin, 211.
— — modifications, 213.
— — problems, 219.
— quoted, 45.
Atmosphere, composition by volume and weight, 63.
Atmospheric pressure, 25.
Atom, 59.
Atomic theory (Dalton), 58.
— weights, 56.
Auxiliary ventilating machines, 173.
— — — axial propulsion type, 174.
— — — peripheral discharge type, 177.
Avogadro's hypothesis, 59.

Backward-bladed fans, 295, 299.
Balancing or synchronising drift for combined fans, 300.
Banking effect of air in fans, 277.
Barclay fan, 287.
Barometer, 25, 31.
— recording, 27.
Beard-Mackie detector, 75.
Beard, Prof., quoted, 216, 225.
Bentley colliery chart, 105.
— — plan, 262.
Bernouilli's theorem, 51.
Bibliography, 37, 62, 91, 123, 147, 208, 249, 270, 289, 311.
Blackdamp, 68.
— tests for, 70.
Blading, 162, 295, 298.
Blowers (fans), 171, 176, 178.
— (gas), Porth, composition of, 71, 72.
Bodily comfort, factors influencing, 117.
Boiling-point, 21, 113.
Bonnets, 260.
Booster fans, characteristic curves, 177.
— — for sinking, 178, 180.
— — the "Jeffrey," 176.
— — the "Schlotter," 175.
Bowhill colliery chart, 107.
Boyle's law, 40.
Brake horse-power, 15.
Brattice sheets, 250.
Bridges, air, 261.
Briggs, Dr., quoted, 81, 100, 117, 143.
— loop, 75.
Brightside fan, 281.
British thermal unit, 23.
Brownlie, David, quoted, 115.
Bryson, Thos., quoted, 130.

Calorie, small, large, and English, 23.
Capell fan, 288, 299.
Caps, for methane-air mixtures with oil flame, 74.
Carbon dioxide, 65, 69.
—monoxide, 69, 81.
— — blood test for, 82.
— — Graham's apparatus, 84.
— — preparation, properties, etc., 81.
Carboxy-hæmoglobin, 82.
Carburetted hydrogen, see Methane.

Casings for fans, 156.
Centigrade, conversion to Fahrenheit, 22.
Centrifugal and centripetal force, 12, 157.
Chandler fan, 274.
Charles' or Dalton's law, 38.
Chemical change, 2.
— compound, 1.
— equivalents, 57.
Chezy quoted, 212.
Chokedamp, 68.
Circular motion, 11.
City Deep Mine, Rand, 93, 107.
Clifford, E. H., quoted, 123.
Clowes' hydrogen lamp, 75.
C.M.A. quoted, 30, 32, 35, 67, 77, 164, 250, 263, 266.
Code of ventilation signs, 251.
Coefficient of friction, 18, 139.
Colliery Guardian quoted, 76.
Comparison of mixtures and compounds, 62.
Conductance of mine and fan, 145, 204.
Conduction, 20.
Conductivity, 20, 100.
Convection, 20.
Cooling effect of emission of gases, 102.
— — of evaporation of moisture, 103.
— — of expansion of air, 102.
— — of vaporisation of water, 102.
— influences in mines, 102.
— mechanical methods of, 108.
— practical mining methods of, 106.
Coward, Dr., quoted, 73.
Creep in driving belts, 193.
Critical pressure, 25, 87.
— temperature, 25, 87.
— volume, 25, 87.
Cunningham-Cadman device, 74.
Cut-off or beak of fan, 43.
Cyclone dust collector, 182.

Dalton's atomic theory, 58.
— or Charles' law, 38.
— law of partial pressures, 47.
Damp; black, fire, stink, and white, 68.
Davies, Eric, quoted, 108, 117.
Definite proportions, 57.
Definitions, 1 *et seq.*
Density, 2.
— of a gas, 60.
— of mine gases and air, 43.
Depression defined, 134.
— effective, 135, 137.
— initial, 135, 136.
— theoretical, 135.
Depression, useless, 135, 137.
Dew point, 30.
Diffusion, law and rates of, 50.
Diffusivity, 20, 100.
Doors, 251.
— notes on, 253.
— self-acting, 256, 257.
Double inlet fans, 158.
Drive efficiency, 188.
Drives for fans, 189.
— — belt or chain, 192.
— — direct vertical steam, 190.
— — — electric motor, 191.
— — — horizontal steam, 191.
— — rope, 192.
Dry bulb, 105, 116.
Dust collector, 182.

Eccentricity of fan in casing, 158.
Effect of carbon monoxide on life, 82.
Effective depression, 137.
Efficiency defined, 15.
— air system, 188
— drive, 188.
— in ventilation, 186.
— manometrical, 187.
— total, 188.
— volumetric, 189.
Electric, miners' lamps, 76.
— motors for fan drives, 192.
Elements, 2.
Emission of gases, cooling effect, 103.
Enclosure of fan, 158.
Energy, conservation of, 7.
— kinetic, potential, pressure, 51.
— K.E. to P.E., 131.
Evaporation of moisture, cooling effect, 103.
Evasée chamber, 160, 161.
Expansion of air, cooling effect, 103.
— of liquid air, 107.

Fahrenheit, conversion to Centigrade, 22.
Fans, centrifugal, 155.
— — backward-bladed, 271.
— — combinations of, 300.
— — common losses in, 103.
— — compound-bladed, 282.
— — drives, 189.
— — effect of dimensions, 299.
— — — of varying resistances, 203.
— — enclosure of, 158.
— — forcing *versus* exhausting, 171.
— — forward-bladed, 275.
— — in parallel, 304.
— — in series, 303.
— — object of, 157.

Fans, centrifugal, position in circuit, 170.
— — practical test of, 204.
— — principle of, 156.
— — radially-bladed, 199, 204, 243, 275.
— — salient features, 155.
— — sizes, 204.
Firedamp, 68, 71.
Fletcher, C., quoted, 257.
Force, 2, 12.
Friction defined, 18.
— factor, 210.
— in air pipes, 134.
— liquid and solid, 18, 19.
Furnaces, 150.
— combined, 152.
— dumb-drift, 151.
— open, 151.
— side-drift, 151.
— thermal calculations, 154.
Fundamental laws and theorems, 38.

Galloway, Dr., quoted, 95.
Gas constant (R), 43.
Gases in atmosphere, 63.
— in mines, 67.
Gassy pillars, 265.
Gauge, recording, 27.
Gay Lussac's law of volumes, 58.
General regulations quoted, 30, 32, 67, 250.
Geothermal gradient, 93, 94.
Gibson, Prof., quoted, 213.
Graham, J. Ivon, quoted, 69, 83, 85, 86, 88, 89, 102.
Graham's law of diffusion, 50.
Guibal fan, 272.

Hæmoglobin, 81, 85.
Halbaum quoted, 225, 229.
Haldane quoted, 71, 73, 77, 82, 84, 98, 104.
Harger quoted, 68.
Harmonic motion in rope drives, 199.
Head, 51.
— air column of, 126, 128, 129, 132, 139, 149, 199, 231, 233, 238.
— conditions for maximum gain of, 291.
— effect of blade curvature on, 295.
— fans in combination, 303.
— gain due to velocity fall, 291.
— loss due to change of direction of air, 290.
— loss in Venturi meter, 55.
— throughout fan, 294.
Heat, combustion and formation of, 25.
Heat, conduction, 19.
— conductivity, 20, 100.
— convection, 20.
— defined, 19.
— diffusivity, 20, 100.
— latent, 24.
— mechanical equivalent of, 23.
— radiation, 20.
— specific, 23, 24, 100.
— unit of, 22.
Henry's law, 49.
Hill, Dr. L., quoted, 117, 122.
— Miss A. M., quoted, 90.
Hodgson, J. L., quoted, 53.
Holman, B. W., quoted, 117, 119.
Homogeneous air column, 30.
Homologous series, 71.
Horse-power, B.H.P., I.H.P., 15, 188.
Humidity, 30, 92, 117.
— absolute, 113.
— principles of, 113.
— relative, 113.
Hydrocarbon gases, 71.
Hydrograph, Briggs', 115.
Hygrometer, care of, 113.
— normal, 31, 104, 113.
— recording, 31.
— Storrow's, 113.
Hygrometric tables, 312.

Indicated horse-power, 15.
Induced draught fans, 179.
Inductance in fans, 277.
Inertia, moment of, 10.
Inherence, 215.
— problems on, 220.
Inherent power "in the air," 215.
— pressure of the airway, 216.
Initial depression, 136.
— — factors influencing, 137.
Inlet and chimney area ratio, 143.
— and orifice ratio, 143.
Innes, *The Fan* quoted, 291.
Iron and Coal Trades Review quoted, 108.
Iron pyrites, effects of mine atmospheres on, 89.
Isothermal conditions, 42.

Jeffrey booster, 176.
— mine fan, 283.
Jets, 150.
Joule, mechanical equivalent of heat, 23.

Kata chart, dry, 121.
— table, wet, 123.
-thermometer, normal, 118.

Kata chart, recording, 119.
Keith-Blackman fan, 168, 277.
Kick's law of crushing, 99.
Kinetic energy of mine-air, 131, 161.

Lamplough, F. E. E., quoted, 90.
Latent heat, 24.
Law, Atkinson's, 54, 209.
— — modifications of, 213.
— — origin of, 211.
— Boyle's, 40.
— Charles' or Dalton's, 38.
— Dalton's (partial pressures), 47.
— definite or constant proportions, 57.
— fans in parallel, 306.
— — in series, 304.
— Gay Lussac's (volumes), 58.
— Graham's (diffusion), 50.
— Henry's (gas solubility), 49, 47.
— Kick's (crushing), 99.
— multiple proportions, 57.
— (s), Newton's (motion), opp. p. 1.
— — — first law quoted, 6.
— — — second law quoted, 17, 296.
— — — third law quoted, 18.
— Petit and Dulong's, 61.
— Rittinger's (crushing), 99.
— Venturi's, 53.
Leakage, 202.
Legal definition of main airway, 266.
— — of ventilating district, 266.
— requirements, see C.M.A. and General Regulations.
Leveing quoted, 76.
Lewes, Prof., quoted, 72.
Limits of explosibility, mine gases with air, 79.

Manometer, 126, 130.
— inclined, 135.
Manometrical efficiency, 187.
Mass, 2.
Matter, 1.
Methane, 57, 71, 72, 77.
— explosive limits of, 77, 79.
— percentage by weight and volume, 77.
— preparation of, 74.
— test flame (oil) caps and percentages, 74.
— tests for, 74.
Mine air, 67, 86.
— — and gases, 68.
— — — outbursts, 80.
— — pressures, 124 *et seq.*
— temperatures and humidity, 92.
Mixtures, 1.
— and compounds compared, 62.
Moisture in air of mines, etc., 102, 103.
Molecules, 59.
Momentum, 15.
— moment of, 16.
Motion, circular, 11.
Motive column, 48, 49, 128, 129, 154.
Multiple proportions, 57.
Murgue quoted, 139.

Natural air-crossings, 262.
— splitting, 222.
— ventilation, 148.
Negative orifice of passage, 144.
Newly opened mines, 155.
Newton's laws of motion, opp. p. 1.
Nitrogen, blackdamp in, 69.
— properties and preparation of, 64.
— solubility in water, 47.
Nominal air-space in fans, 160.
N.T.P. (normal temperature and pressure), 41.
Nystagmus, 85.

Open-running fans, 161, 273.
Ordinary water gauge, 126.
Orifice, and chimney ratio, 143.
— (s), combined, 142.
— effect of varying water gauges on, forcing fan, 176, 203.
— equivalent of mine, 138.
— of passage, 140.
— relation between orifice of mine and fan, 141, 203.
— relation between orifice of mine and fan and inlet, 143.
— series in, 143.
Origin of Atkinson's formula, 211.
O'Shea, Prof., quoted, 46.
Outbursts of gas, 80.
Overcasts, 261, 262.
Oxidation, 97.
Oxygen, solubility in water, 47.
— preparation, etc., 63.

Parker, Jos., quoted, 137, 162, 273, 304, 309.
Peele, Prof., quoted, 143.
Penman, D., quoted, 93, 143, 303, 305.
Peripheral velocity, 135, 199, 210.
Physical change, 2.
— properties of mine gases, 87.
Physiological effects of humid atmospheres, 105.
Pitot tube, 133.
Plans, 263, 267.
— notes on, 268.
— ventilation, 267.
Potential energy, mine air, 131, 161.

Power, 14.
— electric motors, 303.
— laws, 241.
— — application of, 241.
— regulation for fans, 308.
Practice of mine ventilation, 250.
Pressure, 4.
— absolute, 25.
— atmospheric, 25.
— critical, 25.
— laws, 238.
— — application of, 239.
— static, dynamic, and total, 132.
Problems on apportionment, 246.
— on Atkinson's formula, 219.
— on combined fans, 309.
— on fans, 243.
— on inherence, 220.
— on regulators, 229.
— on splits, 227.
Production of ventilation, 148.
Psychrometer, recording, 32.

Quantity, application of laws, 237.
— laws, 236.
— wasted by regulator, 229.
Quick-running fans, 199.
Quiescent air, 131.

Radial air velocity in fans, 298.
Radiation, 20.
Ransom, E. C., quoted, 171.
Rateau fan, 279.
— Prof., quoted, 137.
Reaumur temperature scale, 22.
Recording barometers, 27.
— kata-thermometer, 119.
— psychrometer, 32.
— water-gauge, 127.
Redmayne, Sir R., quoted, 46
Rees, J. R., quoted, 113, 117.
Regnault quoted, 45.
Regulators, 229.
— best position for, 234.
Resistance equivalent of combined fans, 302, 310.
— in mines, 124, 145, 146.
Resultant velocities, 146, 162, 290, 292, 297.
Reunions and splits, 228.
Reversal of ventilation, 164.
— — by converting the fan, 165.
— — conditions for and against, 165.
— — connecting drift, 165.
— — independent blower, 164.
— — open running fan, 166.
— — surface drift, 166.
Reynolds, Prof., quoted, 212.
Rittinger's law of crushing, 99.
"Robbery" in paralleled fans, 300.
Robson, Dr. Fdk., quoted, 85.
Roblings, Geo., quoted, 80.

St. John del Rey Mine, Brazil, 108.
Saturation point, 30.
Siemens-Schuckert fan, 286.
Sinking shaft ventilation, 170.
Sirocco fan, 169, 276.
Slip of driving-belts, 193.
Solubility of gas, or partial pressure, 48.
Solutions, 61.
— decinormal, 62.
— normal, 61.
— standard, 62.
South African Inst. of Engineers quoted, 66.
Speak, Pres. J. S., quoted, 99.
Specific gravity, 3, 61.
— heat, 23.
Speed of fans, 199.
Split problems, 228.
Splits, 222; equal splits, 223.
— formulæ, 223, 226, 227.
Splitting, importance of, 228.
Static head, 133, 176, 295.
Stokes' alcohol lamp, 75.
Stoppings, permanent, 250.
— temporary, 266.
Stythe, 68.
Sulphur in coal, 89.
Surging and mounting in rope drives, 199.
Synchronising or balancing drift for paralleled fans, 300.

Table I. Specific heats of mine gases, 24.
,, II. and III. Methane and barometer, 29.
,, IV. The gas constant (R), 43.
,, V. Atomic weights, 56.
,, VI. Influence of aqueous vapour on weight of air, 66.
,, VII. Limits of explosibility of inflammable gases, 79.
,, VIII. Limits of explosibility of methane-air mixtures, 79.
,, IX. Physical properties of mine gases, 87.
,, X. Geothermal gradient, etc., 94.
,, XI. Thermal characteristics of strata, 100.
,, XII. Wet kata cooling powers, 123.

Table XIII. Direct units of resistance, 146.
„ XIV. Hygrometric data, 312 *et seq.*
Tangential components of velocity in fans, 162, 292, 299.
Temperature, 20, 92.
— absolute, 22.
— atmospheric, 92.
— body, 97.
— critical, 25.
— due to auto-compression in downcast, 95.
— — crushing by subsidence, 99.
— — gob fires, 97.
— — men, animals, etc., 97.
— — natural heat of strata, 93.
— — oxidation, 97.
— mine, 92.
Theoretical depression, 137.
Thermal characteristics of strata, 100.
Thermometers, Centigrade, Fahrenheit, 21.
Thomas, D. E., quoted, 133.
— T. J., quoted, 76.
Time factor in splitting, 228.
Titration, 62, 83.
Torricelli quoted, 26.
Total efficiency, 188.
Trautwine quoted, 67.
Triangular blading, 278, 287.
Turbon fan, 282.
Turbulent fluids, 211.
Turf for cavities and door sealings, 253.
Turquand quoted, 253.

Undercast, 264.
Underground fans, 171.
— fires, 204.
Unit, British thermal, 23.
— of heat, 22.
— power, 223.
— pressure, 223.
— suggested new units, 144, 309.
Unwin, Prof., quoted, 213.
Upcast calculations, 48.
— temperature, furnace, 151.
— top coverings, 255, 258, 260
Useless depression, 137.
Useless effect, 188.
U.S.A. Coast and Geodetic Survey quoted, 46.

Vacuum, partial in fan, 157.
Vaporisation of water, 102.
Vectorial velocities, 146, 162, 290, 292, 297.
Velocity, absolute and relative, 4.
— angular, 10.
— head, 129.
— high, disadvantages of, 202.
— laws, 242.
Ventilating pressure, 124, 128.
Venturi head, 55.
— meter, 53.
Venturi's law, 53.
Virtual slip of air particles in fans, 137.
Volume, critical, 25.
— ratio, 189.
Volumetric efficiency, 189.

Waddle fan, 167, 273.
Walker fan, 164, 273.
Water gauge, 126, 131.
— — depressions, theoretical, initial, effective, etc., 135–138.
— — effect of varying resistances on, 203.
— — head, 132.
— — recording, 127.
— — types, 130.
— vapour, factors influencing, 65, 66, 113.
Weight, 3.
— of air and hydrogen, 3.
— of cubic foot of air, 45, 129.
— of cubic foot of air, dry, 46.
— of mercury, 28.
— of water, 28, 124.
Weighting zones, 268.
Weights, atomic (international table), 56.
Wet bulb, 104, 116.
Wheeler, Dr R. V., quoted, 71, 73.
Whitedamp, 68.
Winmill, T. J., quoted, 85, 88, 90, 91.
Wood, E. Seymour, quoted, 289, 301.
Work, 6.

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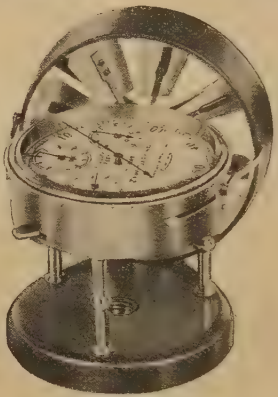

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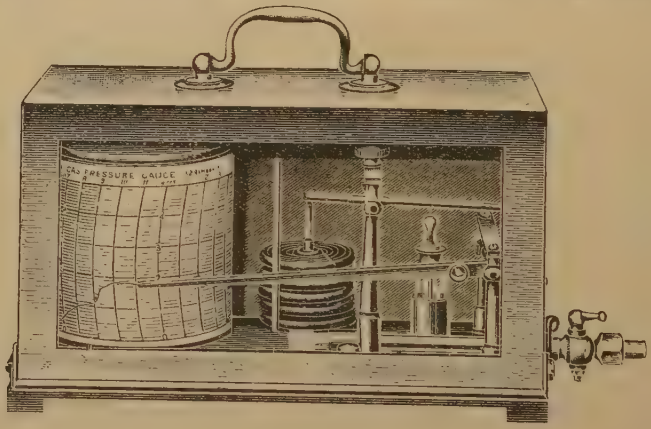

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